Mechanical Engineering

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A POCKET-BOOK

OF

MECHANICAL ENGINEERING

TABLES, DATA, FORMULAS, THEORY
AND EXAMPLES

FOR ENGINEERS AND STUDENTS

BY

CHARLES M. SAMES, B.Sc.
Mechanical Engineer

THIRD EDITION, REVISED AND ENLARGED
THIRD THOUSAND

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PREFACE.

THIS book is the result of the writer's endeavor to compact the greater part of the reference information usually required by mechanical engineers and students into a volume whose dimensions permit of its being carried in the pocket without inconvenience.

In its preparation he has consulted standard treatises and reference books, the transactions of engineering societies, and his own memoranda which extend back over a period of fifteen years. A large amount of valuable and timely matter has been obtained from the columns of technical periodicals and also from the catalogues which manufacturers have courteously placed at his disposition.

While very great care has been taken in the preparation of manuscript and in the reading of proofs, it is nevertheless a regrettable fact that first editions are not always infallible, and the writer will accordingly be under obligations to those who will call his attention to such errors in statement or typography as may come to their notice.

Suggestions indicating how subsequent editions may be made of greater usefulness are respectfully solicited.

CHARLES M. SAMES.

SECOND EDITION, FOR 1907.

ALL matter contained in the first edition has been carefully scrutinized for errors, comparisons having been made with the original sources of the information from which it was compiled, as it was found that nearly all the inaccuracies occurred through recopying from notes.

A number of alterations have been made in the text, certain data have been replaced by fresher matter, and the work has been enlarged by the addition of an appendix in which new subjects are treated, some omissions supplied, and much space given to recent and valuable matter relating particularly to Machine Design.

C. M. S.

THIRD EDITION, FOR 1908.

New matter has been added on a number of subjects, including those of Reinforced Concrete, High-speed Tool Steel, Superheated Steam and Journal Friction. A few changes have also been made in the text, in order to bring it down to the date of publication.

C. M. S.

CONTENTS.

SYMBOLS AND ABBREVIATIONS.

A	Area in square feet.
Am. Mach.	American Machinist.
a	Area in square inches.
Bm	Bending moment.
B.H.P.	Brake horse-power.
B. T.	Board of Trade.
B.T.U.	British thermal unit.
B.W.G.	Birmingham wire gauge.
C.	Centigrade.
C	Modulus of transverse elasticity.
C. I.	Cast iron.
c.	Center.
cm.	Centimeters.
c. g.	Center of gravity.
cir. mils	Circular mils.
c.-p.	Candle-power.
cu.	Cubic.
coeff.	Coefficient.
D	Larger, or outside, diameter in inches.
d	Diameter in inches (diam.).
degs.	Degrees.
E	Modulus of direct elasticity.
E.H.P.	Electrical horse-power.
E.M.F.	Electro-motive force.
E. N.	Engineering News.
E. R.	Engineering Record.
E. W. & E.	Electrical World and Engineer.
F.	Fahrenheit.
Fn	Tractive force in pounds.
f	Acceleration in feet per second.
f_c, f_s, f_t	Stresses in pounds per square inch (compression, shear, tension).
f_r	Modulus of rupture.
ft.	Feet.
ft.-lbs.	Foot-pounds.
G	Pounds in one cubic foot of water.
g	Acceleration of gravity in feet per second (=32.16); Grams.
gal.	Gallons.
g-cal.	Gram-calories.
H	Height or head in feet; total heat in steam above 32° F., in B.T.U.
H.P.	Rated horse-power.
h	Height in inches; sensible heat in the liquid above 32° F.
hor.	Horizontal.
hr.	Hours.
I	Moment of inertia.
I_p	Polar moment of inertia.
I.H.P.	Indicated horse-power.
Ing. Taschenbuch.	Engineer's Pocket Book (Hütte), Berlin.
in.	Inches.

K	Modulus of volumetric elasticity.
k_v	Specific heat at constant volume.
k_p	“ “ “ “ pressure.
kg.	Kilograms; kg.-m., kilogram-meters.
km.	Kilometers.
kw.	Kilowatts.
L	Length in feet; latent heat in B.T.U. per lb. of steam.
l	Length in inches.
lb.	Pounds.
lin.	Linear.
M	Poisson's ratio.
M.E.P.	See *pm*.
M.M.F.	Magneto-motive force.
m	Mass in pounds $= w \div g$.
m.	Meters.
mm.	Millimeters.
m.-kg.	Meter-kilograms.
N	Number of revolutions per minute.
n	“ “ “ “ second.
P	Total pressure in pounds.
p	Pressure, in pounds per square inch.
p''	Pitch, in inches (rivets, screws, gear-teeth).
pm	Mean effective pressure in pounds per square inch.
perp.	Perpendicular.
Q	Flow of air or water in cubic feet per minute.
q	“ “ “ “ “ “ “ “ “ second.
R	Radius in feet; thermodynamic constant.
r	Radius in inches; radius of gyration in inches; ratio of expansion.
r.p.m.	Revolutions per minute.
S	Modulus of section in bending.
S_t	“ “ “ “ torsion.
s	Side of square in inches; distance in feet in velocity formulas.
sec.	Seconds.
sp. gr.	Specific gravity.
sq.	Square.
T	Absolute temperature in degs. F. (also τ).
Tm	Twisting moment.
Tn	Greater tension in belt or rope.
t	Thickness in inches; time in seconds.
t° (or t)	Temperature, or rise of temperature in degs. F.
t_n	Lesser tension in belt or rope.
V	Velocity in feet per minute; volume in cubic feet.
v	Velocity in feet per second.
vert.	Vertical.
W. I.	Wrought iron.
w	Weight or load in pounds (also wt.).
yd.	Yards.
Z. V. D. I.	Zeitschrift des Vereines deutscher Ingenieure. Berlin.
α (Alpha)	Coefficient of linear expansion in degs. F.; an angle.
β (Beta)	An angle.
γ (Gamma)	Pitch angle in spiral gears.
Δ (Delta)	Total deflection in feet; $\Delta_{''}$ = same in inches.
δ “	δ_c, δ_l, δ_s, δ_t Deflection or strain per inch of length (due to compression, laterally, shear, and tension, respectively).
η (Eta)	Efficiency.
θ (Theta)	Angle of torsion.
μ (Mu)	Coefficient of friction; tangent of friction angle.
π (Pi)	Ratio of circumference to diameter $= 3.14159+$.
ρ (Rho)	Radius of curvature in bending.
Σ (Sigma)	Symbol indicating summation.
τ (Tau)	Absolute temperature in degs. F.; normal pitch in spiral gears.
ϕ (Phi)	Entropy.
$\propto$	“Varies as.”
$>$	Greater than.
$<$	Less than.
$\parallel$	Parallel to.
$+$	Across.

MATHEMATICS.

WEIGHTS AND MEASURES (ENGLISH).

Length. 1,000 mils = 1 inch; 12 inches = 1 foot; 3 feet = 1 yard; 5.5 yards = 1 rod, pole or perch; 7.92 inches = 1 link; 100 links = 1 chain; 80 chains = 1 mile = 5,280 feet; 1 furlong = 40 rods; 1 knot or nautical mile = 6,080.26 feet = $\frac{1}{3}$ league.

Surface. 144 sq. in. = 1 sq. ft.; 9 sq. ft. = 1 sq. yd.; 30.25 sq. yd. = 1 sq. rod; 160 sq. rods = 1 acre = 43,560 sq. ft.; 1 circ. mil = 0.0000007854 sq. in.

Volume. 1,728 cu. in. = 1 cu. ft.; 27 cu. ft. = 1 cu. yd.; 1 cord of wood = 128 cu. ft.; 1 perch of masonry = 24.75 cu. ft.

Avoirdupois Weight. (The grain is the same in all systems.) 27.34375 grains = 1 drachm = $\frac{1}{16}$ ounce; 1 pound = 16 oz. = 7,000 grains; 1 long ton = 2,240 lb.; 1 net or short ton = 2,000 lb.

Troy Weight. 24 grains = 1 pennyweight; 20 pennyweights = 1 ounce; 12 ounces = 1 lb. = 5,760 grains; 1 carat = 3.168 grains (= 0.205 gram).

Apothecaries' Weight. 20 grains = 1 scruple; 3 scruples = 1 drachm; 8 drachms = 1 oz.; 12 oz. = 1 lb. = 5,760 grains.

Liquid Measure. 4 gills = 1 pint; 2 pints = 1 quart; 4 quarts = 1 gallon (U. S. gal. = 231 cu. in.; British Imperial gal. = 277.274 cu. in); 31.5 gal. = 1 barrel; 2 barrels = 1 hogshead.

Apothecaries' Fluid Measure. 60 minims = 1 fluid drachm; 8 drachms = 1 fluid ounce = 437.5 grains.

Dry Measure, U. S. 2 pints = 1 quart; 8 quarts = 1 peck; 4 pecks = 1 bushel = 2,150.42 cu. in. = 1.2445 cu. ft. (1 British bushel = 8 Imperial gal. = 2,218.192 cu. in = 1.2837 cu. ft.).

Circular Measure. 60 seconds = 1 minute; 60 minutes = 1 degree; 90 degrees = 1 quadrant = $\frac{1}{4}$ circumference

Board Measure (B. M.). No. of feet board measure = length in feet × width in feet × thickness in inches.

METRIC MEASURES.

The following prefixes are employed for subdivisions and multiples: Milli = 0.001, Centi = 0.01, Deci = 0.1, Deca = 10, Hecto = 100, Kilo = 1,000, Myria = 10,000.

Length. 1 meter = 39.370113 in. = 3.28084 ft. 1 kilometer = 3,280.843 ft. = 0.62137 mile. 1 inch = 2.54 centimeters (cm.) = 25.4 millimeters. 1 foot = 0.3048 meter = 30.48 cm. 1 mile = 1.6093 kilometers = 1609.3 meters.

Surface. 1 square cm. = 100 sq. mm. = 0.155 sq. in. 1 sq. meter (m.) = 10.764 sq. ft. 1 are = 100 sq. m. 1 hectare = 100 ares = 10,000 sq. m. = 2.4711 acres. 1 acre = 0.4047 hectare. 1 sq. mile = 259 hectares. 1 sq. ft. = 0 092903 sq. m. 1 sq. in. = 6.4516 sq. cm.

Volume. 1 stere = 1 kiloliter = 1 cu. meter = 35.3148 cu. ft. 1 liter (l.) = 1 cu. decimeter = 61.024 cu. in. = 0.2642 gal. (U. S.). 1 gal. (U. S.) = 3.7854 liters. 1 cu. cm. = 0.061 cu. in.

Weight. 1 gram (or gramme) = 15.432 grains. 1 kilogram (kg.) = 2.20462 lb. avoirdupois. 1 metric ton = 1,000 kg. = 2,204.62 lb. 1 grain = 0.0648 gram. 1 lb. = 0.4536 kg.

Pressure and Weight. 1 lb. per sq. in. = 0.070308 kg. per sq. cm. 1 kg. per sq. cm. = 14.223 lb. per sq. in. = 1 metric atmosphere. 1 atmosphere (14.7 lb. per sq. in.) = 2,116.3 lb. per sq. ft. = 33.947 ft. of water = 30 in. of mercury (762 mm.) at 62° F. 1 lb. per sq. in. = 27.71 in. of water = 2.0416 in. of mercury at 62° F.

ARITHMETIC AND ALGEBRA.

Squares and Cubes of Numbers. Circumferences and Areas of Circles.

n	n^2	n^3	πn	$\pi n^2 \div 4$
1	1	1	3.142	0.7854
2	4	8	6.283	3.1416
3	9	27	9.425	7.0686
4	16	64	12.566	12.5664
5	25	125	15.708	19.6350
6	36	216	18.850	28.2743
7	49	343	21.991	38.4845
8	64	512	25.133	50.2655
9	81	729	28.274	63.6173
10	100	1000	31.416	78.5398
11	121	1331	34.558	95.0332
12	144	1728	37.699	113.097
13	169	2197	40.841	132.732
14	196	2744	43.982	153.938
15	225	3375	47.124	176.715
16	256	4096	50.265	201.062
17	289	4913	53.407	226.980
18	324	5832	56.549	254.469
19	361	6859	59.690	283.529
20	400	8000	62.832	314.159
21	441	9261	65.973	346.361
22	484	10648	69.115	380.133
23	529	12167	72.257	415.476
24	576	13824	75.398	452.389
25	625	15625	78.540	490.874
26	676	17576	81.681	530.929
27	729	19683	84.823	572.555
28	784	21952	87.965	615.752
29	841	24389	91.106	660.520
30	900	27000	94.248	706.858
31	961	29791	97.389	754.768
32	1024	32768	100.531	804.248
33	1089	35937	103.673	855.299
34	1156	39304	106.814	907.920
35	1225	42875	109.956	962.113
36	1296	46656	113.097	1017.88
37	1369	50653	116.239	1075.21
38	1444	54872	119.381	1134.11
39	1521	59319	122.522	1194.59
40	1600	64000	125.66	1256.64
41	1681	68921	128.81	1320.25
42	1764	74088	131.95	1385.44
43	1849	79507	135.09	1452.20
44	1936	85184	138.23	1520.53
45	2025	91125	141.37	1590.43
46	2116	97336	144.51	1661.90
47	2209	103823	147.65	1734.94
48	2304	110592	150.80	1809.56
49	2401	117649	153.94	1885.74
50	2500	125000	157.08	1963.50
51	2601	132651	160.22	2042.82
52	2704	140608	163.36	2123.72
53	2809	148877	166.50	2206.18
54	2916	157464	169.65	2290.22
55	3025	166375	172.79	2375.83
56	3136	175616	175.93	2463.01
57	3249	185193	179.07	2551.76
58	3364	195112	182.21	2642.08

Squares and Cubes of Numbers. Circumferences and Areas of Circles.

n	n^2	n^3	πn	$\pi n^2 \div 4$
59	3481	205379	185.35	2733.97
60	3600	216000	188.50	2827.43
61	3721	226981	191.64	2922.47
62	3844	238328	194.78	3019.07
63	3969	250047	197.92	3117.25
64	4096	262144	201.06	3216.99
65	4225	274625	204.20	3318.31
66	4356	287496	207.35	3421.19
67	4489	300763	210.49	3525.65
68	4624	314432	213.63	3631.68
69	4761	328509	216.77	3739.28
70	4900	343000	219.91	3848.45
71	5041	357911	223.05	3959.19
72	5184	373248	226.19	4071.50
73	5329	389017	229.34	4185.39
74	5476	405224	232.48	4300.84
75	5625	421875	235.62	4417.86
76	5776	438976	238.76	4536.46
77	5929	456533	241.90	4656.63
78	6084	474552	245.04	4778.36
79	6241	493039	248.19	4901.67
80	6400	512000	251.33	5026.55
81	6561	531441	254.47	5153.00
82	6724	551368	257.61	5281.02
83	6889	571787	260.75	5410.61
84	7056	592704	263.89	5541.77
85	7225	614125	267.04	5674.50
86	7396	636056	270.18	5808.80
87	7569	658503	273.32	5944.68
88	7744	681472	276.46	6082.12
89	7921	704969	279.60	6221.14
90	8100	729000	282.74	6361.73
91	8281	753571	285.88	6503.88
92	8464	778688	289.03	6647.61
93	8649	804357	292.17	6792.91
94	8836	830584	295.31	6939.78
95	9025	857375	298.45	7088.22
96	9216	884736	301.59	7238.23
97	9409	912673	304.73	7389.81
98	9604	941192	307.88	7542.96
99	9801	970299	311.02	7697.69
100	10000	1000000	314.16	7853.98

Square and Cube Root by Approximation. From above table take n whose cube or square is nearest the number of which the root is desired. For square root, divide the number by n, obtaining the quotient n_1; take $(n+n_1)\div 2$ $(=n_2)$ for a new divisor, obtaining n_3 as a quotient; take $(n_2+n_3)\div 2$ for a new divisor and continue process until divisor and quotient are alike, or to the required accuracy.

For cube root, divide the number by n^2, obtaining quotient n_1; take $\left(\frac{2n+n_1}{3}\right)^2 = n_2{}^2$ for a new divisor, obtaining quotient n_3; take $\left(\frac{2n_2+n_3}{3}\right)^2$ for a new divisor and continue process until $(2nx+nx_{+1})\div 3=$ quotient.

Compound Interest. $a=c(1+p)^n$, where $a=$ amount, $c=$ initial capital, $p=$ rate per cent in hundredths, and $n=$ number of years.

Binomial Theorem.

$$(a\pm b)^n=a^n\pm na^{n-1}b+\frac{n(n-1)}{1.2}a^{n-2}b^2\pm\frac{n(n-1)(n-2)}{1.2.3}a^{n-3}b^3+\ \dots$$

Arithmetical and Geometrical Progression. Let a=first term of the series, b=last term, d=difference between any two adjacent terms (in Arith. Prog.), n=number of terms, s=sum of all the terms, r=ratio of any term divided by preceding one (in Geom. Prog.). Then, for Arithmetical series, $b=a+(n-1)d=\frac{2s}{n}-a$;

$$s=\frac{n}{2}[2a+(n-1)d]=\frac{b+a}{2}+\frac{b^2-a^2}{2d}=(b+a)\frac{n}{2}=\frac{n}{2}[2b-(n-1)d].$$

For Geometrical series, $b=ar^{n-1}=\frac{a+(r-1)s}{r}=\frac{(r-1)sr^{n-1}}{r^n-1}$;

$$s=\frac{a(r^n-1)}{r-1}=\frac{rb-a}{r-1}=\frac{b(r^n-1)}{(r-1)r^{n-1}}=\frac{\sqrt[n-1]{b^n}-\sqrt[n-1]{a^n}}{\sqrt[n-1]{b}-\sqrt[n-1]{a}};\quad n=1+\frac{\log b-\log a}{\log r}$$

Sinking Fund for Depreciation and Renewal. $s=a(r^n-1)\div(r-1)$, where s is the fund or amount to be accumulated in n years, and $r=1$ plus the rate per cent of interest to be compounded annually, the rate being expressed in hundredths. Example. A certain machine costing \$1,000 (s) will need to be replaced by a new one costing the same amount at the end of 10 years (n). What sum must be paid into a sinking fund at the end of each year to amount to \$1,000 at the end of the tenth year, interest being compounded at the rate of 5 per cent? $1{,}000=a(1.05^{10}-1)\div(1.05-1)$, and a, or the annual amount to be placed in the fund, =\$79.50.

Interpolation. Where a value intermediate to two values in a table is desired, the following formula may be employed. Value desired,

$$a_x=a+nb+\frac{n(n-1)c}{1.2}+\frac{n(n-1)(n-2)d}{1.2.3}+\ldots$$

Let N, N_1, N_2 and N_3 be four numbers (equally spaced) whose tabular functions are a, a_1, a_2 and a_3. Then, in above formula to find a_x, the tabular function of N_x (lying between N and N_1), $n=\frac{N_x-N}{N_1-N}$.

b=the first of the first order of differences,
c= " " " " second " " "
d= " " " " third " " " , etc.

Example. The chords of 30°, 32°, 34° and 36° are 0.5176, 0.5513, 0.5847 and 0.6180, respectively. Find the chord of 31°.

	a		a_1		a_2		a_3
	0.5176		0.5513		0.5847		0.6180
$b=$ 0.0337		0.0337		0.0334		0.0333	
$c=$ −0.0003			−0.0003		−0.0001		
$d=$ 0.0002				0.0002			

$$n=(31-30)\div(32-30)=0.5$$

$$a_x=0.5176+0.5(0.0337)+\frac{0.5(-0.5)(-0.0003)}{2}+0.5\frac{(-0.5)(-1.5)(0.0002)}{6}$$
$=0.5345.$

Logarithms (log). The hyperbolic or Napierian log of any number equals the common log×2.3025851. The common log of any number equals the hyperbolic log ($\log_e$)×0.4342945.

Every log consists of a whole part (the characteristic) and a decimal part (the mantissa). The mantissa or decimal part only is given in the tables.

The characteristic of the log of a number is one less than the number of figures to the left of the decimal point in the number.

Log 3=.47712, log 30=1.47712, log 300=2.47712, etc

Log 0.3= −1.47712, log 0.03= −2.47712, log 0.003= −3.47712, etc.

Any logarithm with a negative characteristic as −1.47712, may be written as 9.47712−10. (The sum of 9 and −10 being −1.)

Formulas for Using Logarithms. $\log ab=\log a+\log b$.

$$\log\frac{a}{b}=\log a-\log b.\quad \log a^b=b\log a.\quad \log\sqrt[b]{a}=\frac{\log a}{b}.$$

Examples.

5×4 (using logs); Log 5 = .69897
Log 4 = .60206

Sum = 1.30103, which is the log of 20, or the result required.

Multiply 0.5 by 0.04.

log 0.5 = −1.69897 = 9.69897 − 10
log 0.04 = −2.60206 = 8.60206 − 10

Their sum = 18.30103 − 20 = −2.30103, or the log of 0.02.

For 0.5 ÷ 0.04, diff. of logs = 1.09691 − 0 = log of 12.5.

Find nth root of 0.09.

log 0.09 = −2.95424 = 8.95424 − 10
divided by n (say 2) = 4.47712 − 5 = −1.47712, or log of 0.3.

Raise 0.3 to nth power.

log 0.3 = −1.47712 = 9.47712 − 10
multiplying by n (say 2) = 18.95424 − 20 = −2.95424 = log 0.09.

Log $\pi = .49715$, $\log \frac{1}{\pi} = -1.50285$, $\log \pi^2 = .9943$,

$\log \sqrt{\pi} = .248575$. $\pi = 3.1415926536+$.

TABLE OF CHORDS.

Deg.	Chd.	Deg.	Chd.	Deg.	Chd.	Deg.	Chd.	Deg.	Chd.
2	.0349	20	.3473	38	.6511	56	.9389	74	1.2036
4	.0698	22	.3816	40	.6840	58	.9700	76	1.2313
6	.1047	24	.4158	42	.7167	60	1.0000	78	1.2586
8	.1395	26	.4499	44	.7492	62	1.0301	80	1.2856
10	.1743	28	.4838	46	.7815	64	1.0598	82	1.3121
12	.2090	30	.5176	48	.8135	66	1.0893	84	1.3383
14	.2437	32	.5513	50	.8452	68	1.1184	86	1.3640
16	.2783	34	.5847	52	.8767	70	1.1471	88	1.3893
18	.3129	36	.6180	54	.9080	72	1.1756	90	1.4142

MENSURATION.

AREAS OF PLANE FIGURES (A).

Triangles. Take as base any side which will be intersected by a perpendicular let fall from vertex of opposite angle. Length of base $= b$, length of side to the left $= a$, side to right $= c$. Then $A = \frac{b}{2}\sqrt{a^2 - \left(\frac{a^2+b^2-c^2}{2b}\right)^2} = bh \div 2$, where h = length of perpendicular.

Trapezoid. If a, b and h = lengths of parallel sides and perpendicular, respectively, $A = 0.5h(a+b)$.

Circle. (r = radius, d = diameter) $A = \pi r^2 = \pi d^2 \div 4$. Circumf. $= \pi d$.

Sector of Circle. $A = 0.5r \times$ length of arc $= 0.008727 r^2 \times$ degrees in arc.

Segment of Circle. $A = 0.5[br - c(r-h)]$. b = arc, c = base, h = height at center of base.

Ellipse. Equation referred to axes through center: $a^2y^2 + b^2x^2 = a^2b^2$, where a = semi-minor axis, b = semi-major axis and x and y are the abscissa and ordinate of any point on the perimeter. $A = \pi ab$. Length of perimeter $= \pi(a+b)\left[1 + \frac{1}{4}\left(\frac{a-b}{a+b}\right)^2 + \frac{1}{64}\left(\frac{a-b}{a+b}\right)^4 + \frac{1}{256}\left(\frac{a-b}{a+b}\right)^6 \ldots\right]$

Parabola. Equation, origin at vertex: $y^2 = 2px$, where $2p$ is the parameter, or double ordinate through focus. Area of any portion from vertex $= \frac{2xy}{3}$.

Hyperbola. Equation: $a^2y^2 - b^2x^2 = -a^2b^2$.

Cycloid. Length of curve = 4 times diam. of generating circle.
Area = 3 " area " "

Area of Any Irregular Figure. Simpson's Rule. Divide the length of the figure into an even number of equal parts and erect ordinates through the points of division to touch the boundary lines. Then $A = \left(\frac{a + 4b + 2c}{3}\right)d$, where a = sum of first and last ordinates, b = sum of even ordinates, c = sum of odd ordinates (excepting first and last) and d = common distance between ordinates. The greater the number of divisions the greater will be the accuracy.

LOGARITHMS OF NUMBERS.

No.	0	1	2	3	4	5	6	7	8	9	Diff.
10	00000	00432	00860	01284	01703	02119	02531	02938	03342	03743	415
11	04139	04532	04922	05308	05690	06070	06446	06819	07188	07555	379
12	07918	08279	08636	08991	09342	09691	10037	10380	10721	11059	344
13	11394	11727	12057	12385	12710	13033	13354	13672	13988	14301	323
14	14613	14922	15229	15534	15836	16137	16435	16732	17026	17319	298
15	17609	17898	18184	18469	18752	19033	19312	19590	19866	20140	281
16	20412	20683	20952	21219	21484	21748	22011	22272	22531	22789	264
17	23045	23300	23553	23805	24055	24304	24551	24797	25042	25285	249
18	25527	25768	26007	26245	26482	26717	26951	27184	27416	27646	234
19	27875	28103	28330	28556	28780	29003	29226	29447	29667	29885	222
20	30103	30320	30535	30750	30963	31175	31387	31597	31806	32015	212
21	32222	32428	32634	32838	33041	33244	33445	33646	33846	34044	202
22	34242	34439	34635	34830	35025	35218	35411	35603	35793	35984	193
23	36173	36361	36549	36736	36922	37107	37291	37475	37658	37840	185
24	38021	38202	38382	38561	38739	38917	39094	39270	39445	39620	177
25	39794	39967	40140	40312	40483	40654	40824	40993	41162	41330	170
26	41497	41664	41830	41996	42160	42325	42488	42651	42813	42975	164
27	43136	43297	43457	43616	43775	43933	44091	44248	44404	44560	158
28	44716	44871	45025	45179	45332	45484	45637	45788	45939	46090	153
29	46240	46389	46538	46687	46835	46982	47129	47276	47422	47567	148
30	47712	47857	48001	48144	48287	48430	48572	48714	48855	48996	143
31	49136	49276	49415	49554	49693	49831	49969	50106	50243	50379	138
32	50515	50651	50786	50920	51055	51189	51322	51455	51587	51720	134
33	51851	51983	52114	52244	52375	52504	52634	52763	52892	53020	130
34	53148	53275	53403	53529	53656	53782	53908	54033	54158	54283	126
35	54407	54531	54654	54777	54900	55023	55145	55267	55388	55509	122
36	55630	55751	55871	55991	56110	56229	56348	56467	56585	56703	119
37	56820	56937	57054	57171	57287	57403	57519	57634	57749	57864	116
38	57978	58093	58206	58320	58433	58546	58659	58771	58883	58995	113
39	59106	59218	59329	59439	59550	59660	59770	59879	59988	60097	110
40	60206	60314	60423	60531	60638	60746	60853	60959	61066	61172	107
41	61278	61384	61490	61595	61700	61805	61909	62014	62118	62221	104
42	62325	62428	62531	62634	62737	62839	62941	63043	63144	63246	102
43	63347	63448	63548	63649	63749	63849	63949	64048	64147	64246	99
44	64345	64444	64542	64640	64738	64836	64933	65031	65128	65225	98
45	65321	65418	65514	65610	65706	65801	65896	65992	66087	66181	96
46	66276	66370	66464	66558	66652	66745	66839	66932	67025	67117	95
47	67210	67302	67394	67486	67578	67669	67761	67852	67943	68034	92
48	68124	68215	68305	68395	68485	68574	68664	68753	68842	68931	90
49	69020	69108	69197	69285	69373	69461	69548	69636	69723	69810	88
50	69897	69984	70070	70157	70243	70329	70415	70501	70586	70672	86
51	70757	70842	70927	71012	71096	71181	71265	71349	71433	71517	84
52	71600	71684	71767	71850	71933	72016	72099	72181	72263	72346	82
53	72428	72509	72591	72673	72754	72835	72916	72997	73078	73159	81
54	73239	73320	73400	73480	73560	73640	73719	73799	73878	73957	80

LOGARITHMS OF NUMBERS (*Continued*).

No.	0	1	2	3	4	5	6	7	8	9	Diff.
55	74036	74115	74194	74273	74351	74429	74507	74586	74663	74741	78
56	74819	74896	74974	75051	75128	75205	75282	75358	75435	75511	77
57	75587	75664	75740	75815	75891	75967	76042	76118	76193	76268	75
58	76343	76418	76492	76567	76641	76716	76790	76864	76938	77012	74
59	77085	77159	77232	77305	77379	77452	77525	77597	77670	77743	73
60	77815	77887	77960	78032	78104	78176	78247	78319	78390	78462	72
61	78533	78604	78675	78746	78817	78888	78958	79029	79099	79169	71
62	79239	79309	79379	79449	79518	79588	79657	79727	79796	79865	70
63	79934	80003	80072	80140	80209	80277	80346	80414	80482	80550	69
64	80618	80686	80754	80821	80889	80956	81023	81090	81158	81224	68
65	81291	81358	81425	81491	81558	81624	81690	81757	81823	81889	67
66	81954	82020	82086	82151	82217	82282	82347	82413	82478	82543	66
67	82607	82672	82737	82802	82866	82930	82995	83059	83123	83187	64
68	83251	83315	83378	83442	83506	83569	83632	83696	83759	83822	63
69	83885	83948	84011	84073	84136	84198	84261	84323	84386	84448	63
70	84510	84572	84634	84696	84757	84819	84880	84942	85003	85065	62
71	85126	85187	85248	85309	85370	85431	85491	85552	85612	85673	61
72	85733	85794	85854	85914	85974	86034	86094	86153	86213	86273	60
73	86332	86392	86451	86510	86570	86629	86688	86747	86806	86864	59
74	86923	86982	87040	87099	87157	87216	87274	87332	87390	87448	58
75	87506	87564	87622	87680	87737	87795	87852	87910	87967	88024	57
76	88081	88138	88196	88252	88309	88366	88423	88480	88536	88593	57
77	88649	88705	88762	88818	88874	88930	88986	89042	89098	89154	56
78	89209	89265	89321	89376	89432	89487	89542	89597	89653	89708	55
79	89763	89818	89873	89927	89982	90037	90091	90146	90200	90255	54
80	90309	90363	90417	90472	90526	90580	90634	90687	90741	90795	54
81	90849	90902	90956	91009	91062	91116	91169	91222	91275	91328	53
82	91381	91434	91487	91540	91593	91645	91698	91751	91803	91855	53
83	91908	91960	92012	92065	92117	92169	92221	92273	92324	92376	52
84	92428	92480	92531	92583	92634	92686	92737	92788	92840	92891	51
85	92942	92993	93044	93095	93146	93197	93247	93298	93349	93399	51
86	93450	93500	93551	93601	93651	93702	93752	93802	93852	93902	50
87	93952	94002	94052	94101	94151	94201	94250	94300	94349	94399	49
88	94448	94498	94547	94596	94645	94694	94743	94792	94841	94890	49
89	94939	94988	95036	95085	95134	95182	95231	95279	95328	95376	48
90	95424	95472	95521	95569	95617	95665	95713	95761	95809	95856	48
91	95904	95952	95999	96047	96095	96142	96190	96237	96284	96332	48
92	96379	96426	96473	96520	96567	96614	95661	96708	96755	96802	47
93	96848	96895	96942	96988	97035	97081	97128	97174	97220	97267	47
94	97313	97359	97405	97451	97497	97543	97589	97635	97681	97727	46
95	97772	97818	97864	97909	97955	98000	98046	98091	98137	98182	46
96	98227	98272	98318	98363	98408	98453	98498	98543	98588	98632	45
97	98677	98722	98767	98811	98856	98900	98945	98989	99034	99078	45
98	99123	99167	99211	99255	99300	99344	99388	99432	99476	99520	44
99	99564	99607	99651	99695	99739	99782	99826	99870	99913	99957	44

Note.—The differences in the last column are mean values only. For accurate values the difference between any two consecutive values should be found by subtraction.

SURFACES (A) AND VOLUMES (V) OF SOLIDS.

Sphere. $A=4\pi r^2=\pi d^2$. $V=\frac{\pi d^3}{6}=0.5236d^3$.

Ring of Circular Cross-section. $A=9.8696Dd$. $V=2.4674Dd^2$. ($D=$ outside diameter $-d$; $d=$diam. of cross-section.)

Segment of Sphere. $A = 2\pi rh =$ area of base $+ \pi h^2$ ($h =$ height).

$$V = \pi h^2\left(r - \frac{h}{3}\right).$$

Cone. $A = \pi r\sqrt{r^2 + h^2}$. $V = 0.2618 d^2 h$ ($h =$ vert. height).

Conic Frustum. $A = \frac{\pi}{2}(D + d) \times$ slant height, h.

$$V = \frac{\pi h}{12}(D^2 + Dd + d^2).$$

Cylinder. $V = 0.7854 d^2 h$. (d is the revolving axis of cyl. and ellipsoid.)
Ellipsoid. $V = 0.5236 Dd^2$. **Paraboloid.** $V = 1.5708 r^2 h$.

Pyramid. $V = \frac{h}{3} \times$ area of base.

Frustum of Pyramid. $V = \frac{h}{3}(A + a + \sqrt{Aa})$ (A and $a =$ areas of bases).

TRIGONOMETRY.

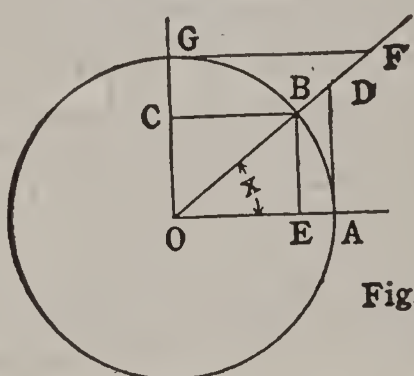

Fig. 1.

Functions of the angle $BOE (= x)$. $EB =$ sine, $OE =$ cosine, $EA =$ versed sine, $GC =$ versed cosine, $AD =$ tangent, $GF =$ cotangent, $OD =$ secant, $OF =$ cosecant.

Formulas. (A, B and C are angles.)

$\tan A = \frac{\sin A}{\cos A}$; $\cot A = \frac{\cos A}{\sin A}$; $\sec A = \frac{1}{\cos A}$; $\operatorname{cosec} A = \frac{1}{\sin A}$; $\tan A = \frac{1}{\cot A}$.

$\sin^2 A + \cos^2 A = 1$; $\operatorname{versin} A = 1 - \cos A$; $\operatorname{covers} A = 1 - \sin A$.

$\sin(A \pm B) = \sin A \cos B \pm \cos A \sin B$.

$\cos(A \pm B) = \cos A \cos B \mp \sin A \sin B$.

$\sin A = 2 \sin \frac{A}{2} \cos \frac{A}{2}$. $\cos A = \cos^2 \frac{A}{2} - \sin^2 \frac{A}{2}$.

$1 - \cos A = 2 \sin^2 \frac{A}{2}$. $1 + \cos A = 2 \cos^2 \frac{A}{2}$.

$\tan A = 2 \tan \frac{A}{2} \div \left[1 - \tan^2 \frac{A}{2}\right]$. $\sin A + \cos A = \sin\left(\frac{\pi}{4} + A\right)\sqrt{2}$.

$\cos A - \sin A = \sin\left(\frac{\pi}{4} - A\right)\sqrt{2}$. $\frac{1 - \cos A}{\cos A} = \tan A \tan \frac{A}{2}$.

$\tan(A \pm B) = [\tan A \pm \tan B] \div [1 \mp \tan A \tan B]$.

$\cot(A \pm B) = [\cot A \cot B \mp 1] \div [\cot A \pm \cot B]$.

$\sin A \pm \sin B = 2 \sin \frac{A \pm B}{2} \cos \frac{A \mp B}{2}$.

$\cos A + \cos B = 2 \cos \frac{A + B}{2} \cos \frac{A - B}{2}$.

$\cos A - \cos B = -2 \sin \frac{A + B}{2} \sin \frac{A - B}{2}$.

$\sin A \sin B = \frac{1}{2} \cos(A - B) - \frac{1}{2} \cos(A + B)$.

$\cos A \cos B = \frac{1}{2} \cos(A + B) + \frac{1}{2} \cos(A - B)$.

$\sin A \cos B = \frac{1}{2} \sin(A + B) + \frac{1}{2} \sin(A - B)$.

$\sin 3A = 3 \sin A - 4 \sin^3 A$. $\cos 3A = 4 \cos^3 A - 3 \cos A$.

$(\cos A \pm i \sin A)^n = \cos nA \pm i \sin nA$ ($i = \sqrt{-1}$).

If $A+B+C=180°=\pi$ (the three angles of a triangle), then

$$\sin A+\sin B+\sin C=4\cos\frac{A}{2}\cos\frac{B}{2}\cos\frac{C}{2}.$$

$$\cos A+\cos B+\cos C=1+4\sin\frac{A}{2}\sin\frac{B}{2}\sin\frac{C}{2}.$$

$$\tan A+\tan B+\tan C=\tan A\tan B\tan C.$$

NATURAL TRIGONOMETRICAL FUNCTIONS.

Degs.	Sine.	Tangent.		Degs.	Sine.	Tangent.	
0	.00000	.00000	90	46	.71934	1.03553	44
1	.01745	.01746	89	47	.73135	1 07237	43
2	.03490	.03492	88	48	.74314	1.11061	42
3	.05234	.05241	87	49	.75471	1.15037	41
4	.06976	.06993	86	50	.76604	1.19175	40
5	.08716	.08749	85	51	.77715	1.23490	39
6	.10453	.10510	84	52	.78801	1.27994	38
7	.12187	.12278	83	53	.79864	1.32704	37
8	.13917	.14054	82	54	.80902	1.37638	36
9	.15643	.15838	81	55	.81915	1.42815	35
10	.17365	.17633	80	56	.82904	1.48256	34
11	.19081	.19438	79	57	.83867	1.53987	33
12	.20791	.21256	78	58	.84805	1.60033	32
13	.22495	.23087	77	59	.85717	1.66428	31
14	.24192	.24933	76	60	.86603	1.73205	30
15	.25882	.26795	75	61	.87462	1.80405	29
16	.27564	.28675	74	62	.88295	1.88073	28
17	.29237	.30573	73	63	.89101	1.96261	27
18	.30902	.32492	72	64	.89879	2.05030	26
19	.32557	.34433	71	65	.90631	2.14451	25
20	.34202	.36397	70	66	.91355	2.24604	24
21	.35837	.38386	69	67	.92050	2.35585	23
22	.37461	.40403	68	68	.92718	2.47509	22
23	.39073	.42447	67	69	.93358	2.60509	21
24	.40674	.44523	66	70	.93969	2.74748	20
25	.42262	46631	65	71	.94552	2.90421	19
26	.43837	.48773	64	72	.95106	3.07768	18
27	.45399	.50952	63	73	.95630	3.27085	17
28	.46947	.53171	62	74	.96126	3.48741	16
29	.48481	.55431	61	75	.96593	3.73205	15
30	.50000	.57735	60	76	.97030	4.01078	14
31	.51504	.60086	59	77	.97437	4.33148	13
32	.52992	.62487	58	78	.97815	4.70463	12
33	.54464	.64941	57	79	.98163	5.14455	11
34	.55919	.67451	56	80	.98481	5.67128	10
35	.57358	.70021	55	81	.98769	6.31375	9
36	.58779	.72654	54	82	.99027	7.11537	8
37	.60182	.75355	53	83	.99255	8.14435	7
38	.61566	.78129	52	84	.99452	9.51436	6
39	.62932	.80978	51	85	.99619	11.43005	5
40	.64279	.83910	50	86	.99756	14.30067	4
41	.65606	.86929	49	87	.99863	19.08114	3
42	.66913	.90040	48	88	.99939	28.63625	2
43	.68200	.93252	47	89	.99985	57.28996	1
44	.69466	.96569	46	90	1.00000	Infinite	0
45	.70711	1.00000	45				
	Cosine.	Cotangent.	Degs.		Cosine.	Cotangent.	Degs.

For intermediate values reduce angles from degrees, minutes and seconds to degrees and decimal part of a degree (e.g., 46° 21′ 30″ = 46.3583°) and employ interpolation formula.

CHEMICAL DATA.

Atomic Weights and Symbols of Elements.

Aluminum	Al	26.9	Molybdenum	Mo	95.3
Antimony	Sb	119.3	Neodymium	Ne	142.5
Argon	A	39.6	Neon		19.9
Arsenic	As	74.4	Nickel	Ni	58.3
Barium	Ba	136.4	Nitrogen	N	13.93
Bismuth	Bi	206.9	Osmium	Os	189.6
Boron	B	10.9	Oxygen	O	15.88
Bromine	Br	79.36	Palladium	Pd	106.7
Cadmium	Cd	111.6	Phosphorus	P	30.77
Cæsium	Cs	132	Platinum	Pt	193.3
Calcium	Ca	39.8	Potassium	K	'38.86
Carbon	C	11.91	Praseodymium	Pr	139.4
Cerium	Ce	139	Radium	Ra	223.3
Chlorine	Cl	35.18	Rhodium	Rh	102.2
Chromium	Cr	51.7	Rubidium	Rb	84.8
Cobalt	Co	58.56	Ruthenium	Ru	100.9
Columbium (Niobium)	Cb	93.3	Samarium	Sm	148.9
Copper	Cu	63.1	Scandium	Sc	43.8
Erbium	E	164.8	Selenium	Se	78.6
Fluorine	F	18.9	Silicon	Si	28.2
Gadolinium	Gd	155	Silver	Ag	107.12
Gallium	Ga	69.5	Sodium	Na	22.88
Germanium	Ge	71.9	Strontium	Sr	86.94
Glucinum (Beryllium)	Gl	9.03	Sulphur	S	31.83
Gold	Au	195.7	Tantalum	Ta	181.6
Helium	He	4	Tellurium	Te	126.6
Hydrogen	H	1.00	Terbium	Tb	158.8
Indium	In	113.1	Thalium	Tl	202.6
Iodine	I	125.9	Thorium	Th	230.8
Iridium	Ir	191.5	Thulium	Tm	169.7
Iron	Fe	55.5	Tin	Sn	118.1
Krypton	K	81.2	Titanium	Ti	47.7
Lanthanum	La	137.9	Tungsten	W	182.6
Lead	Pb	205.35	Uranium	U	236.7
Lithium	Li	6.98	Vanadium	V	50.8
Magnesium	Mg	24.18	Xenon	X	127
Manganese	Mn	54.6	Ytterbium	Yb	171.7
Mercury	Hg	198.5	Yttrium	Yt	88.3
			Zinc	Zn	64.9
			Zirconium	Zr	89.9

Calculation of the Percentage Composition of Substances. (1) Add together the atomic weights of the elements to obtain the molecular weight of the compound. (2) Multiply the atomic weight of the element to be calculated by the number of atoms present (as indicated by the subscript number) and by 100, and divide by the molecular weight of the compound.

Example. Find the percentage of sulphur in sulphuric acid (H_2SO_4).

$H_2 + S + O_4$

$(1\times2)+31.83+(15.88\times4)=97.35$, or the molecular weight. $3183 \div 97.35 = 32.59$, or the percentage of sulphur in the acid.

Weights of Gases. Avogadro's law: "In equal volumes of all gases there are the same number of molecules." It follows from this law that the weights of equal volumes of all gases are proportional to their molecular weights.

The molecular or formula weight in grams of any gas occupies 22.4 liters at 0° C. and 760 mm. pressure.

Example. Find the weight of one liter of carbon dioxide (CO_2). Molecular wt. of $CO_2=11.91+(15.88\times2)=43.67$. $\therefore$ 43.67 grams = 22.4 liters, or 1 liter weighs 1.95 grams.

(1 cu. ft. = 28.317 liters; 1 liter = 0.03532 cu. ft.; 1 lb. = 453.5924 grams; 1 gram = 0.0022046 lb.)

MATERIALS.

Cast Iron (C. I.). Sp. gr. = 7.21; wt. per cu. in. = 0.261 lb. Fusing point of white iron = 1,962° F.;—gray iron, 2,192° F. Chemically composed of iron (Fe), carbon (C) (graphitic and combined), silicon (Si), phosphorus (P), sulphur (S) and manganese (Mn). Contains 3.5 to 4% of total carbon, the hardness of castings varying directly with the amount of combined carbon. Si (from 0.5 to 3.5%) produces softness and strength proportional to amount contained. (Best at 1.8%.) S beyond 0.15% is prejudicial, producing blow-holes and brittleness when hot. P promotes fluidity but causes brittleness when in excess of 1%. Mn assists the carbon in combining and confers the property of chilling. It should not exceed 1%.

Wrought Iron (W. I.). Sp. gr. = 7.78; wt. per cu. in. = 0.282 lb. Consists of over 99% pure iron + 0.3% combined carbon + 0.14% each of S, Si and P.

Steel. Cast steel, sp. gr. = 7.92; wt. per cu. in. = 0.286 lb. Forged steel, sp. gr. = 7 82; wt. per cu. in. = 0.283 lb. Fusing point = 2.500 to 2,700° F.

Temper (or content of carbon). Castings, 0.3 to 0.4%; forgings, 0.25 to 0.3%: chains, 0.15 to 0.18%; laminated springs, 0.4 to 0.6%; boiler plates, 0.17 to 0.2%; same, for welding, 0.15 to 0.17%; tool steel, up to 1.35%.

Manganese Steel (containing 14% Mn) has double the strength of ordinary steel combined with great hardness.

Nickel Steel (3 to 5% Ni) has 30% greater tenacity and 75% greater elastic strength than ordinary mild steel, along with equal ductility. Harveyized, for ship armor, it offers the same resistance with 43% less weight.

Chrome Steel (0.4% C + 1% of Chromium (Cr) + 2% Ni) is of extreme hardness (self-hardening) and is used for safe walls, projectiles, and cutting tools.

Tungsten Steel (Mushet's) is a self-hardening steel for tools, shells, etc. (1.70% C + 0.42% Si + 0.25% Mn + 8.5% Tungsten (W)).

Copper (Cu). Sp. gr. = 8 878 (wire and rolled); wt. per cu. in. = 0.321 lb.; fusing point = 1,950° F. **Zinc** (Zn). Sp. gr. = 6.86 (cast); wt. per cu. in. = 0 248 lb.; fusing point = 787° F. **Tin** (Sn). Sp. gr. = 7 3; wt. per cu. in. = 0.264 lb; fusing point = 446° F. **Aluminum** (Al). Sp. gr. = 2 56 (cast) and 2.68 (rolled); wt. per cu. in. = 0.092 lb. (cast) and 0.097 lb. (rolled). Fuses at 1,213° F.

Mercury (Hg). Sp. gr = 13.619 (at 32° F) and 13.58 (at 60° F.); wt. per cu. in. = 0.493 lb. (at 32° F) and 0.491 lb. (at 60° F.). Fuses at −39° F.

Gun Metal Bronze (80 to 90% Cu + 20 to 10% Sn) Strong and tough Increasing the content of tin increases the hardness **Phosphor Bronze** (85% Cu + 15% Sn + 0 5 to 0.75% P) has the toughness of W I. **Manganese Bronze** (81% Cu + 12% Sn + 7% Mn) is even stronger. **Silicon Bronze** (Cu + 3 to 5% Si) has a breaking stress of 55,000 to 75,000 lb. per sq. in., but at and around 5% Si, is brittle. **Aluminum Bronze** (Cu + 5 to 11% Al) has a slightly greater strength. **Brass** (60 to 70% Cu + 40 to 30% Zn). **Babbitt** (89.3% Sn + 3.6% Cu + 7.1% Sb (antimony)).

Alloys. (E. A. Lewis, Engineering, 3-31-05.)

	Cu.	Sn.	Zn.	Pb.	P.	Si.
For steam or gas pressure....	87	9	2	2		
" hydraulic pressure......	86	12	2			
" bearings...............	84	8		8		
Phosphor-bronze...........	84	14		2	0.05	
Copper castings...........	99.75					0.25

Delta Metal (92.4% Cu+2.38% Sn+5.2% Pb (lead)).
Magnolia Metal (83.55% Pb+16.45% Sn). **Tobin Bronze** (59% Cu+2.16% Sn+0.3% Pb+38.4% Zn). **Solder.** 2 Sn+1 Pb fuses at 340° F., 1 Sn+2 Pb fuses at 441° F., and 20 Sn+1 Pb (for aluminum) at 550° F.

Woods. Average Sp. Gr. and Weights per Cu. Ft.

	Sp. Gr.	Wt.		Sp. Gr.	Wt.		Sp. Gr.	Wt.
Ash........	0.72	45	Fir........	0.59	37	Red Oak...	0.74	46
Beech......	.73	46	Hickory....	.77	48	White Pine.	.45	28
Birch......	.65	41	Hemlock...	.38	24	Yellow Pine.	.61	38
Cedar......	.62	39	Maple......	.68	42	Poplar.....	.48	30
Elm........	.61	38	White Oak..	.77	48	Spruce.....	.45	28

Stones and Miscellaneous Building Materials.

	Sp. Gr.	Wt.		Sp. Gr.	Wt.
Asbestos.............	3.07	192	Graphite.....	2.16	135
Asphaltum...........	1.39	87	Glass........	2.64–2.93	164–183
Brick (com.).........	1.6	100	Limestone....	2.7–3.2	170–200
" (pressed)........	2.16	135	Marble.......	2.56–2.88	160–180
" (fire)...........	2.24	140	Mica.........	2.8	173
Clay.................	1.92	120	Quartz.......	2.64	165
Cement, Rosendale....	0.96	60	Rubber......	0.933	58.4
" Portland......	1.25	78	Sand.........	1.9	122
Earth (loose).........	1.28	80	Sandstone....	2.4	150
Granite..............	2.6	165	Slate........	2.88	180

(Wts. in lbs. per cu. ft.)

Weight of Rods, Bars, Plates, Tubes, and Spheres of Metals.

Material.	Lbs. per cu. ft.	Square Bars, lbs. per lin. ft.	Flat Bars, lbs. per lin. ft.	Round Rods, lbs. per lin. ft.	Plates, lbs. per sq. ft.	Spheres, lbs.
Cast Iron......	450	$3.125s^2$	(Substitute bt for s^2 in preceding column.)	$2.454d^2$	$37.5t$	$0.1363d^3$
Wrought Iron..	480	$3.333s^2$		$2.618d^2$	$40t$	$0.1455d^3$
Steel..........	489.6	$3.4s^2$		$2.670d^2$	$40.8t$	$0.1484d^3$
Copper........	552	$3.833s^2$		$3.010d^2$	$46t$	$0.1673d^3$
Brass (65 Cu+35 Zn)......	523.2	$3.633s^2$		$2.853d^2$	$43.6t$	$0.1586d^3$
Aluminum......	166.5	$1.156s^2$		$0.908d^2$	$13.875t$	$0.0504d^3$

For tubes, multiply numerical coeff. for round rods by $(d^2-d_1^2)$.
For hollow spheres, multiply numerical coeff. for spheres by $(d^3-d_1^3)$.
s=side of square, b=breadth, t=thickness, d=external diam., d_1=internal diam., all in inches.

Weight of Square and Round Wrought Iron Bars in Lbs. per Lineal Foot.

s or d.	Rd.	Sq.	s or d.	Rd.	Sq.	s or d.	Rd.	Sq.
1/16	.010	.013	11/16	1.237	1.576	1 5/8	6.913	8.802
1/8	.041	.052	3/4	1.473	1.875	1 3/4	8.018	10.21
3/16	.092	.117	13/16	1.728	2.201	1 7/8	9.204	11.72
1/4	.164	.208	7/8	2.004	2.552	2	10.47	13.33
5/16	.256	.326	15/16	2.301	2.930	2 1/4	13.25	16.88
3/8	.368	.469	1	2.618	3.333	2 1/2	16.36	20.83
7/16	.501	.638	1 1/8	3.313	4.219	2 3/4	19 8	25.21
1/2	.654	.833	1 1/4	4.091	5.208	3	23.56	30
9/16	.828	1.055	1 3/8	4.95	6.302	3 1/2	32.07	40.83
5/8	1.023	1.302	1 1/2	5.89	7.5	4	41.89	53.33

(s=side of sq. in in. d=diam. in in.)

Weight of Flat W. I. Bars (1 in. wide) in Lbs. per Lineal Foot.

Thickness.	Lbs.	Thickness.	Lbs.	Thickness.	Lbs.
$\frac{1}{16}$	.208	$\frac{7}{16}$	1.46	$\frac{3}{4}$	2.50
$\frac{1}{8}$	.417	$\frac{1}{2}$	1.67	$\frac{13}{16}$	2.71
$\frac{3}{16}$	.625	$\frac{9}{16}$	1.88	$\frac{7}{8}$	2.92
$\frac{1}{4}$	.833	$\frac{5}{8}$	2.08	$\frac{15}{16}$	3.13
$\frac{5}{16}$	1.04	$\frac{11}{16}$	2.29	1	3.33
$\frac{3}{8}$	1.25				

Thickness in in. For steel add 2%.

Weight of Iron, Steel, Copper and Brass Sheets per Square Foot.

Lbs. per sq. ft. = thickness in inches (obtained from gauge tables) ×40, 40.8, 46, or 43.6 respectively.

Corrugated and Flat Iron. Lbs. per Sq. Ft.

Thickness in in.	Flat, lbs.	Corr., lbs.	Thickness in in.	Flat, lbs.	Corr., lbs.
.065	2.61	3.28	.028	1.12	1.41
.049	1.97	2.48	.022	0.88	1.11
.035	1.4	1.76	.018	0.72	0.91

If galvanized, add 0.34 lb. per sq. ft. for flat plates and 0.43 lb. for corrugated plates. End laps 4 in. and 6 in. Side laps = 1 corrugation = 2.5 in.

Tin Plates. (Tinned sheet steel.) Usual roofing sizes are 14×20 and 20×28 (in inches). No. 29 B. W. G. weighs 49.6 lb. per 100 sq. ft.; No. 27 weighs 62 lbs. per 100 sq. ft.

Roofing Slate. (1 cu. ft. weighs 175 lb.)

Thickness in in.	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$
Lbs. per sq. ft.	1.81	2.71	3.62	5.43	7.25	9.06	10.88

Slates are generally laid so that the third slate overlaps the first by 3 in. Sq. in. of roof covered by 1 slate $=0.5b(l-3)$. No. of slates required for 1 square (100 sq. ft.) $=28{,}800 \div b(l-3)$. (b and l are breadth and length in in.) Sizes: 6 to 9×12, 7 to 10×14, 8 to 10×16, 9 to 12×8, 10 to 16×20, 12 to 14×22, 12 to 16×24, 14 to 16×26. (Increases by steps of 1 in.)

Pine Shingles. No. per 100 sq. ft. = 3,600 ÷ no. of inches exposed to weather. Wt. in lbs. of 100 sq. ft. = 864 ÷ no. of inches exposed to weather.

Skylight and Floor Glass. Lbs. per sq. ft. = 13 × thickness in inches.

Flagging. Wt. in lbs. per sq. ft. = 14 × thickness in inches.

Approximate Weights of Roofing Materials. (Lbs. per 100 sq. ft.) 1 in. sheathing: spruce, 200; northern yellow pine, 300; southern yellow pine, 400; chestnut and maple, 400; ash and oak, 500. Shingles, 200; $\frac{1}{4}$ in. slate, 900; $\frac{1}{16}$ in. sheet iron, 300; do., with lath, 500; corrugated iron, 100–375; galvanized flat, 100–350; tin, 70–125; felt and asphalt, 100; felt and gravel, 800–1,000; skylights (glass $\frac{3}{16}$–$\frac{1}{2}$), 250–700; sheet lead, 500–800; copper, 80–125; zinc, 100–200; flat tiles, 1,500–2,000; do., with mortar, 2,000–3,000; pan tiles, 1,000.

Weight of Cast-iron Pipe per Lineal Foot. Wt. in lbs. $=9.81t(d+t)$, where d and t are the internal diam. and thickness of metal in in. The wt. of the two flanges = wt. of 1 ft. of pipe. For copper, multiply by 1.226; for W. I., by 1.067.

Weight of Cast-iron Water and Gas Pipes per Lineal Foot.

Size in in.	4	8	12	16	20	24	30	36	42	48	60
Water, lbs. per ft.	22	42	75	125	200	250	350	475	600	775	1330
Gas, " " "	17	40	70	100	150	184	250	350	383	542	900

Thickness of Cast-iron Water Pipes.

$$t=0.00006(h+230)d+0.333-0.0033d,$$

where h = head of water in feet, and d are thickness and diam. in in.

Riveted Hydraulic Pipe. (Pelton Water Wheel Co.) Head in feet that pipe will safely stand $=48{,}000t \div d$. Weight in lbs. per lin. ft. $=cdt$. $c=15$ for 4 in. pipe 14 up to 8 in. pipe, 13 up to 12 in., 12.5 up to 24 in. and 12 up to 42 in. pipe.

Wrought-iron Pipe Dimensions and Threads. U. S. Standard.

Internal Diam. Nominal in in.	Internal Diam. Actual in in.	Thickness in in.	Lbs. per lin. ft.	Threads per in.	Internal Diam. Nominal in in.	Internal Diam. Actual in in.	Thickness in in.	Lbs. per lin. ft.	Threads per in.
1/8	.270	.068	.24	27	4½	4.508	.246	12.49	8
1/4	.364	.088	.42	18	5	5.045	.259	14.50	8
3/8	.494	.091	.56	18	6	6.065	.28	18.76	8
1/2	.623	.109	.84	14	7	7.023	.301	23.27	8
3/4	.824	.113	1.12	14	8	7.982	.322	28.18	8
1	1.048	.134	1.67	11.5	9	9.001	.344	33.70	8
1¼	1.38	.140	2.24	11.5	10	10.019	.366	40	8
1½	1.611	.145	2.68	11.5	11	11.	.375	45	8
2	2.067	.154	3.61	11.5	12	12.	.375	49	8
2½	2.468	.204	5.74	8	13	13.25	.375	54	8
3	3.067	.217	7.54	8	14	14.25	.375	58	8
3½	3.548	.226	9.	8	15	15.25	.375	.62	8
4	4.026	.237	10.66	8					

Standard Boiler Tubes. Lap-welded Charcoal Iron. (Morris Tasker & Co.)

Outside diam. in.	Inside diam. in.	Lbs. per ft.	Outside diam. in.	Inside diam. in.	Lbs. per ft.
1	0.856	0.708	3½	3.262	4.272
1/4	1.106	0.900	3/4	3.512	4.59
1/2	1.334	1.25	4	3.741	5.32
3/4	1.56	1.665	1/2	4.241	6.01
2	1.804	1.981	5	4.72	7.226
1/4	2.054	2.238	6	5.699	9.346
1/2	2.283	2.755	7	6.657	12.435
3/4	2.533	3.045	8	7.636	15.109
3	2.783	3.333	9	8.615	18.002
1/4	3.012	3.958	10	9.573	22.19

Surface of tube 1 ft. long in sq. ft. $= 0.2618 \times$ diam. in in.

Wrought-iron Welded Tubes. Extra Strong.

Nominal diam. in.	Actual Diameters in in. Outside.	Inside, Ex. Strong.	Inside, Double Ex. Strong.
1/8	0.405	0.205	
1/4	0.54	0.294	
3/8	0.675	0.421	
1/2	0.84	0.542	0.244
3/4	1.05	0.736	0.422
1	1.315	.951	0.587
1/4	1.66	1.272	0.884
1/2	1.9	1.494	1.088
2	2.375	1.933	1.491
1/2	2.875	2.315	1.755
3	3.5	2.892	2.284
1/2	4.	3.358	2.716
4	4.5	3.818	3.136

Lead Pipe. Safe working pressure in lbs. per sq. in. $= 1.000t \div d$. Approx. wt. in lbs. per ft. $= 15\ 5t(\text{caliber} + t)$. t (thickness) and d (diam.) in in.

Number of Square and Hexagonal Nuts in 100 lbs. (U. S. Standard; chamfered, trimmed and punched for standard taps.)

Bolt diam. in in.	No. Sq.	No. Hex.	Bolt diam. in in.	No. Sq.	No. Hex.	Bolt diam. in in.	No. Sq.	No. Hex.
1/4	7270	7615	7/8	280	309	1 3/4	34	40
3/8	2350	3000	1	170	216	2	23	29
1/2	1120	1430	1/8	130	148	1/4	19	21
5/8	640	740	1/4	96	111	1/2	12	15
3/4	380	450	1/2	58	68	3/4	9	11
						3	7.33	8.5

Bolts. Approximate Weight per Hundred. Weight of 100 bolts in lbs. $= a + (b \times \text{length in in.})$.

Bolt diam.	1/4	3/8	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2
Sq. heads and nuts.											
a	=2	5.7	11	23	39	63.6	97	105	190	230	325
b	=1.4	3	5.6	8.4	12.2	16.6	22	30	35	40	50
Hex. heads and nuts.											
a	=1.2	3.7	7	16	27	48	64	66	150	180	250
b	=1.4	3	5.6	8.4	12.2	16.6	22	30	35	40	50

Bridge Rivets. Weight per 100. Weight of 100 rivets in lbs. $= a +$ $(b \times \text{length under head in in.})$.

Diam. in in.	3/8	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4
a	=1.8	5.8	11.1	13.8	22.7	38.8	58.1	83.6
b	=3.13	5.55	8.7	12.5	17	22.25	28.15	34.8

Track Spikes. Number in Keg of 200 Lbs.

Size. .	4½×½	5×7/16	5×½	5×9/16	5½×½	5½×9/16	6×9/16	6×5/8
No. . .	533	650	520	393	466	384	350	260

Wire Nails and Spikes. Number in One Pound.

Size.	Length in.	Common nail.	Barbed.	Fine.	Finishing.	Barbed roof.	Spikes.
2*d*	1	1200	876	1550	1350	411	
4*d*	1½	432	357	760	584	165	
6*d*	2	252	204	350	310	103	
8*d*	2½	132	99	190	170		
10*d*	3	87	69	137	121		50
16*d*	3½	51	43		72		35
20*d*	4	35	31		54		26
30*d*	4½	27	24		46		20
40*d*	5	21	18		36		15
50*d*	5½	15					12
60*d*	6	12					10

Spikes: 6½ in., 9; 7 in., 7; 8 in., 5; 9 in., 4½.

Lag Screws. Approximate Weight per Hundred. Weight of 100 lag screws in lbs. $= a + (b \times \text{length in in.})$.

Diam. in in.	3/8	7/16	1/2	5/8	3/4
a.	2.2	5.7	8	12	24
b.	2.9	3.3	4.6	7.2	10

Iron Wire. Tensile Strength per Square Inch of Section.

Diam. in in.	0.05	0.1	0.2	0.3	0.4
Strength in lbs.	106,000	97,500	87,500	81,000	79,000

The above for bright, charcoal iron wire. If annealed take 75% of values. For Bessemer steel add 10% and for crucible steel 15%.

Galvanized Iron Wire. Weight and Resistance per Mile.
(Roebling.)

B. & S. gauge.	Lbs.	Ohms.	B. & S. gauge.	Lbs.	Ohms.	B. & S. gauge.	Lbs.	Ohms.
6	550	10	9	330	16.4	12	170	32.7
7	470	12.1	10	268	20	13	100	52.8
8	385	14.1	11	216	26	14	62	91.6

Galvanized Steel-wire Strand (7 wires twisted). (Roebling.)

Diam. of rope, in. . . .	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{1}{8}$
Wire gauge No.	8	10	11	12	15	17	20
Lbs. per 100 ft.	52	36	29	21	10	6	2.4

Estimated breaking strength in lbs. = 160 × wt. in lbs. of 100 ft.

Wire Hoisting Rope. (Roebling.) Made from $\frac{1}{4}$ to $2\frac{3}{4}$ in. diam., 6 strands of 19 wires each, hemp center. Wt. in lbs. per ft. $= 1.58d^2$. Approx. breaking strain in lbs. $= cd^2$.

Diam. in in., $d=$	1.5	1	0.5
Swedish iron, $c=$	30,000	32,000	35,000
Cast steel, $c=$	60,000	64,000	70,000

Transmission or Haulage Rope. $\frac{9}{32}$ to $1\frac{1}{2}$ in. in diam., 6 strands of 7 wires each, hemp center.

Diam. in in., $d=$	1.5	1	0.5
Swedish iron, $c=$	30,000	32,000	33,500
Cast steel, $c=$	60,000	64,000	67,000

Extra Strong Crucible Cast-steel Rope (6 strand, hemp center).

Diam. in in., $d=$	2.5	1.5	1	0.5
19 wire strand, $c=$	70,000	75,000	78,000	81,000
7 " " $c=$		70,000	75,000	78,000

Crane Chains (Pencoyd). Pitch in in. (c. of 1 link to c. of next),

$$p'' = 0.17 + 2.43d \quad (\text{where } d < 1\tfrac{1}{4} \text{ in.});$$
$$= 2.75d - 0.156 \quad (\text{ " } \quad d > 1\tfrac{1}{4} \text{ in.});$$

d = diam. of link wire in ins. Outside width of link $= 3.3d + \frac{1}{16}$ in. approx. Approx. wt. per ft. in lbs. for $d = \frac{1}{4}$ to $\frac{1}{2}$ in., wt. $= 0.875 + 6.5(d - \frac{1}{4})$; for $d = \frac{1}{2}$ to $\frac{7}{8}$ in., wt. $= 2.5 + 14.6(d - \frac{1}{2})$; for $d = \frac{7}{8}$ to $1\frac{1}{2}$, wt. $= 8 + 21.9(d - \frac{7}{8})$.

D B G Special Chain. Average breaking strain in lbs. $= 62{,}000d^2$, when $d \leq \frac{3}{4}$ in., and $62{,}000d^2 - 6{,}800(d - \frac{5}{8})$, when $d > \frac{3}{4}$ in. For proof test take $\frac{1}{2}$ of these values, and for safe load $\frac{1}{3}$. Ordinary crane chains have from 87 to 90% of the strength of the D B G special chains. Chain sheaves should have a diameter of not less than $70d$.

Holding Power of Nails and Spikes. (Approximate.) Force in lbs. required to withdraw nail $= csl$, where l = length of nail in the wood in in., and s = circumference of a round nail or the four sides of cut nail in in.

	Values of c. White Pine.	Yellow Pine.	White Oak.
Wrought spikes,	$c=$ 360		720
Wire nails,	$c=$ 167	318	940
Cut nails,	$c=$ 405	662	1216

Weight of Floors. Solid brick arched floors, 70 lbs. per sq. ft. Hollow brick arched floors, from 20 lbs. per sq. ft. for a 3 ft. span to 60 lbs. for a 10 ft. span. Wooden floors, lbs. per sq. ft. per inch of thickness: White Oak, 4; Maple, 3.5; Yellow Pine, 3.2; White Pine and Spruce, 2.33; Hemlock, 2.

Floor Loads in lbs. per sq. ft. Street bridges, 80; dwellings, 40; churches, theatres and assembly rooms, 80; grain elevators, 100; warehouses, 250; factories, 200 to 400. Prof. L. J. Johnson states as the result of experiments that the excessive crowding of adults may produce a load as high as 160 lbs. per sq. ft. 1 cu. ft. of brickwork gives a load of 115 lbs. per sq. ft. of supporting floor. (Masonry, 160 lbs.)

Roof Loads in lbs. per sq. ft. Corrugated iron, 37 to 40; slate, 43 to 46 (add 10 lbs. if plastered below rafters). These values include an allowance of 30 lbs. for wind and snow. Snow per ft. depth, 6.4; maximum wind pressure, 50.

Brick Masonry. Common bricks are $8\frac{3}{4}$ in. $\times 4\frac{1}{2}$ in. $\times 2\frac{3}{4}$ in. Pressed, $8\frac{1}{2}$ in. $\times 4\frac{1}{3}$ in. $\times 2\frac{3}{8}$ in. Wt., 5 to 6 lbs. Number of bricks per sq. ft. of wall surface $= 1.55 \times$ thickness of wall in inches (approx.). 1,000 closely stacked bricks occupy about 56 cu. ft. Safe load for brickwork in tons per sq. ft.: for good lime mortar, 8 tons; for good cement mortar, 15 tons. (N. Y. City Law.)

THE STRENGTH OF MATERIALS, STRUCTURES, AND MACHINE PARTS.

Stress is the cohesive force within the material which is called into action to resist the load or externally applied force.

Strain is the deformation produced by the stress and is proportional to the stress within the elastic limit.

Elasticity is the property which a body possesses of regaining its original shape and dimensions after distortion.

Modulus of Direct Elasticity. $E = \frac{f_t}{\delta_t} = \frac{f_c}{\delta_c}$.

Modulus of Transverse Elasticity. $C = f_s \div \delta_s$ (for shear).

Modulus of Volumetric Elasticity. $K = f_v \div$ decrease in vol. per cu. in.

Elastic Moduli (inch and pound units).

Material.	E	C	K
Cast Steel	30,000,000	12,000,000	26,000,000
Forged Steel	30,000,000	13,000,000	26,000,000
Tempered Steel	36,000,000	14,000,000	
W. I. Bars	29,000,000	10,500,000	20,000,000
" Plates	26,000,000	14,000,000	20,000,000
Copper	12,000,000		24,000,000
" rolled	15,000,000 (for drawn, $E = 17,000,000$)		
Cast Iron	17,000,000	6,300,000	14,000,000
Brass and Gun Metal	13,500,000		15,000,000
Water			300,000

Poisson's Ratio (M). If a bar be extended or compressed, the direct strain (δ_t or δ_c) = lateral strain (δ_l) $\times M$. The value of M for steel is 3.25, for W. I., 3.6, for C. I., 3.7, for copper, 2.6, and for brass, 3.

Work. The unit of work is one foot-pound. Work = pressure or force $\times$ distance = pounds $\times$ feet = ft.-lbs., and may be represented by the area of a figure with abscissæ of distance and ordinates of pressure or force.

Resilience = the work done in deforming a body up to the elastic limit = $\frac{F}{2} \times \Delta_{\prime}$ ft.-lbs. = $\frac{\text{total stress in lbs.}}{2} \times$ deflection in feet.

Stress Due to Impulsive Load. Make energy equal to the resilience. Then, $\frac{wv^2}{2g} = \frac{F\Delta_{\prime}}{2}$, and F (lbs.) $= \frac{wv^2}{g\Delta_{\prime}}$, which is the maximum. The mean total stress (between 0 and max.) $= \frac{wv^2}{2g\Delta_{\prime}}$, which applies to steam-hammers, pile-drivers, etc. In case of a falling weight (e.g., sudden load on a beam or crane chain), $w(h + \Delta_{\prime\prime}) = \frac{F\Delta_{\prime\prime}}{2}$.

Stress Caused by Heat. $F = E\alpha t^{\circ}a$.

Coefficients of Linear Expansion (α) per Deg. F.

Material	α	Material	α
Tempered Steel	.0000073	Cast Iron	.0000062
Strong Steel	.0000063	Brass	.0000105
Mild Steel	.0000057	Copper	.0000095
Wrought Iron	.0000066	Bronze	.0000111

Relative Hardness of Materials. Cast steel, 554; brass, 233; mild steel, 143; aluminum (cast), 103; copper (annealed), 62; zinc (cast), 41; lead, 4. **Strength is increased** as the temperature is lowered, -50 to 100% at $-295°$ F. Iron and steel gain slightly in strength up to 500° F., but thereafter the decrease is rapid.

Factors of Safety.

Safe Load = Breaking Load ÷ Factor of Safety.

	Dead Load.	Live Load.*	Moving and Reversible † Loads.
W. I and Mild Steel . . .	3	5 to 8	9 to 13
Hard Steel.	3	5 to 8	10 to 15
Bronzes.	5	6 to 9	10 to 15
C. I. and Brass.	4	6 to 10	10 to 15
Timber } In permanent structures.		10	
Brickwork } In permanent structures.		6	
Masonry } In permanent structures.		20 to 30	

Herr Wöhler's experiments in 1871 showed that range of variation in stress was a factor in lowering the breaking load and also that rupture may be caused by repetitions and repeated reversals of stress, none of which attain the elastic limit. Prof. Unwin gives the following equation: $f_1=\frac{S}{2}+\sqrt{f^2-xSf}$, where f_1 = the breaking stress under variation, in tons per sq. in., S = stress variation in terms of f_1. $x=1.5$ for W. I. and mild steel and 2 for hard steel, and f = breaking load under steady stress. $S=\frac{\text{highest stress}-\text{lowest stress}}{\text{highest stress}}\times f_1$.

For a steady load $f_1=f$; for a simple live or suddenly applied load, $S=f_1$; for alternately equal tensile and compressive stresses as in shafting, $S=2f_1$, whence, for

	W. I.	Steel.
Steady load.	$f_1=f$	$f_1=f$
Live load.	$f_1=0.6f$	$f_1=0.472f$
Reversible load.	$f_1=0.33f$	$f_1=0.25f$

Or, safety factors are in the ratio 1: 2: 3 to 4, approx.

Average Breaking Stresses of Building Materials.

(In lbs. per sq. in.)

Material.	Tension.	Compression.
White Oak.	10,000 (‖ to grain)	4,500 (columns < 15 × diam.)
" Pine.	7,000 " " "	3,500 " " "
La. Long-leaf Pine.	12,000 " " "	5,000
Hemp Rope.	8,000	
Granite.	600	15,000
Limestone.	1,000	7,000
Sandstone.	150	5,000
Stonework.	(0.4 × strength of stone used)	
Brickwork	50	1,000 (common, in lime mortar)
"	300	2,000 (best, in cement)
Portland Cement, 1 mo. .	400	2,000
" " 1 year.	500	3,000
" Concrete, 1 mo. .	200	1,000
" " 1 year	400	2,000

Rosendale Cement has about ½ the strength of Portland.
Safe strengths of stone, brick, and cement = 0.1 × breaking strengths.

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* A load on and off continually and instantly, but without velocity.
† A reversible load causes alternate tension and compression.

Average Breaking Stresses of Materials and Safe Stresses for Ordinary Live Loads. (In lbs. per sq. in.)

Metals.	Tension.		Compression.		Shear.	
	Breaking.	Safe.	Breaking.	Safe.	Breaking.	Safe.
Crucible Cast Steel.	100,000	18,000	180,000	18,000		11,200
Mild Steel.	78,000	15,500	. . . .	15,500	. . . .	11,200
Structural Steel, 0.1% Carbon. . . .	56,000	11,200	56,000	11,200	48,000	9,000
Do., 0.15% C.	64,000	12,800	. . . .	. . . .	50,000	10,000
Soft Steel.	52–62,000	15,000	(America	n Bridge	Webs =	9,000
Medium Steel.	60–70,000	17,000	Co. Pract	ice.)	" =	10,000
Steel Castings.	67,000	11,200	. . . .	11,200	. . . .	7,800
Iron Forgings.	56,000	11,200	50,000	9,000	45,000	7,800
W. I. Plates ||	50,000	9,000	. . . .	9,000	36,000	6,700
" " + ...	40,000	9,000	. . . .	9,000	36,000	6,700
Cast Iron.	17,000	2,800	100,000	9,000	11,000	2,200
Malleable Iron. . . .	35,000	6,000				
Manganese Steel. . .	135,000	22,500	(14% Mn	)		
Nickel Steel.	83,000	17,000	(Plates)			
" "	100,000	18,000	(Forging	s)		
Manganese Bronze	67,000	11,200	. . . .	11,200	. . . .	7,800
Phosphor Bronze. .	56,000	9,000	. . . .	9,000	. . . .	6,700
Silicon Bronze. . . .	63,000	11,200				
Aluminum Bronze.	67,000	11,200				
Delta Metal.	67,000	11,200	(Forging	s)		
" "	45,000	7,800	(Cast)			
Gun Metal.	27,000	4,500	. . . .	4,500	. . . .	3,360
Copper.	29,000	4,500	58,000	4,500	25,000	3,360
Brass.	25,000	3,360	. . . .	3,360	. . . .	2,200
Copper Wire.	36,000	. . . .	(anneale	d)		
" "	60,000	. . . .	(unanne	aled)		
Iron "	60,000	. . . .	(anneale	d)		
" "	80,000	. . . .	(unanne	aled)		
Steel "	120,000	. . . .	"			
" "	80,000	. . . .	(anneale	d)		
" "	180,000	. . . .	(crucible	steel)		
" "	200,000	. . . .	(bridge c	able)		

Note. Where vacancies occur in table, assume compression to equal tension, and shear to be 0.7 × tension. || means parallel with grain or fiber, + means across grain.

Tensile Stress-Action. Load = Total Stress, or $w = f_t a$, ($= p \times$ area pressed upon in case of steam, air, or water pressure).

Strength of Chain. $w = 14{,}000d^2$ lbs. for safe loading, where d = diam. in in. of the wire in link. Wt. per ft. $= 10d^2$, approx. (See Crane Chains, *ante.*)

Strength of Ropes. w (safe) $= 1{,}120d^2$ for White Hemp. For wire rope, w (safe) $= 20{,}000nd^2$ lbs., where n = no. of wires and d = diam. of wire in in. (See Wire Rope, *ante.*)

Strength of Pipes and Cylinders Pressed Internally.

Thin Cylinders. For a longitudinal section (e.g., boiler) $f_t = \frac{pr}{t}$, and for a transverse or ring section, $f_t = \frac{pr}{2t}$. Stresses f_t must be multiplied by η in the case of boilers or other cylinders where welded, riveted, or bolted construction is used. In this case η = efficiency = strength of joint ÷ strength of solid plate. For ordinary steam, water, or gas pres-

sures, $t=0.18\sqrt{d}$ for pipes and rough cylinders. For machining, in the case of cylinders, add 0.3 in. to above value of t. Kent states as an average derived from a number of rules: $t=0.0004dp+0.3$ in.

Thick Cylinders. (For very high pressures, e.g., hydraulic.) External diam. = Internal diam. $\times\sqrt{f_t+p}\div\sqrt{f_t-p}$.

Tensile Stress induced by Centrifugal Force. $f_t=\frac{12wv^2}{g}$. For cast iron $w=0.261$ lb. and f_t safe = 2,800 lb. Placing these values in formula, v is found to be 170 ft. per sec., or the safe theoretical velocity of a fly-wheel rim (double actual practice).

Strength of Bolts. The working stress per sq. in. of cross-section at the bottom of thread for ordinary joints = 8,000 lbs. for W. I., and 11,000 lbs. for mild steel. (If under steam or water pressure, 6,000 lbs. In this case bolts $<\frac{3}{4}$ in. should not be used and the pitch should not exceed $6d$.) For steam cylinders, etc., No. of bolts $=\frac{p}{2400}\left(\frac{\text{cyl. diam.}}{\text{bolt diam.}}\right)^2$. Where bolts have to resist shock the shanks should be turned down to the diam. at bottom of thread.

Compressive Stress-Action. $w=f_ca$. (Applicable where length $<12d$.) (See Columns.)

Shear Stress-Action. For pins and rivets, $w=f_sa$. f_s safe = 11,000 lbs. per sq. in. (Am. Bridge Co. practice.)

Strength of Eye Bars. f_t safe = 14,000 to 16,000 lb. for soft and medium steel respectively.

Proportions: $D-d=1.4b$; $d=\frac{7}{8}$ to $1\frac{1}{4}b$; t (for $b<5$ in.) = 0.75 in.; t (for $b>5$ in.) $=(b+1)\div 8$ (in.) Radius of fillet at neck $=D=$ outside diam. (Passaic R. M. Co.) $b=d=0.4D$. Fillet radius $=D$. (Shaler Smith.)

Strength of Riveted Joints.—Single-riveted Lap Joint. Shear strength of one rivet = tensile strength of plate between two holes, or $f_s\pi d^2\div 4=f_t(p''-d)t$ (1). d (of rivet) $=1.2\sqrt{t}$ before riveting; $d=d_1$ (of hole) $=1.3\sqrt{t}$ after riveting (for plates ≤ 1 in.). Substituting in (1) and making $f_s=11{,}200$, $f_t=13{,}500$, pitch, $p''=1.09+d_1$ for steel. For iron plates and rivets $p''=1.14+d_1$; for steel plates and iron rivets, $p''=0.76+d_1$; for copper plates and rivets $p=0.98+d_1$. (Supplee gives as standard practice (up to $\frac{1}{2}$ in. plates) 1.31 and 1.25 in place of 1.14 and 0.76 as above.) Center of rivet to edge of plate $=\frac{1}{2}$ overlap $=1.5d$.

Double-riveted Lap Joint (staggered or zigzag). $p''=2.18+d_1$. Distance between rows of rivets $=\sqrt{1.09d_1+0.75d_1^2}$.

Chain-riveted Lap Joint (double riveted, but not staggered). $p''=2.18+d_1$. Distance between rows $=1.5+d_1$.

Double-riveted Butt Joint (with two cover plates). $p''=4.36+d_1$. Diagonal distance between centers of rivets in the two rows $=2.18+d_1$. Thickness of each butt strap or cover plate $=\frac{5}{8}t$ of plate. Overlap $=2d$.

Treble-riveted Butt Joint. This case calls for three rows of rivets. The pitch of the third row from edge is twice the pitch of the first two rows, which are staggered Examining as a lap joint the metal between two holes on pitch line $=(p''-d)=\frac{0.655d^2}{t}=$ the strength of one rivet. As 5 rivets have to be taken care of, then $p''=\frac{3.275d_1^2}{t}+d_1$. Considered as a butt joint, $(p''-d)=\frac{1.31d^2}{t}$, and for 5 rivets, $p''=\frac{6.55d_1^2}{t}+d_1$. An intermediate value is generally taken. ($p''=$ pitch of third row from edge of plate.) In the above formulas p'' is taken equal to d_1 plus 2.18, 4.36, etc., which are multiples of 1.09 in formula for single-riveted lap joint, and are for steel plates and rivets where $\frac{f_s}{f_t}=\frac{11{,}200}{13{,}500}$. For other metals or combinations similar multiples of 1.14, 0.76, 0.98, etc., should be used, or, if other safe stresses are chosen for f_s and f_t, values of p'' should be worked out from formula (1). Overlap $=1\frac{3}{4}$ to $2d$ for treble-riveted butt joint, thickness of butt strap $=\frac{5}{8}t$ of plate.

Rivet Proportions. Round or snap head: large diam. $=1.67\times$ rivet

diam, and height of head $=\frac{5}{8}d$. Countersunk head: large diam. $=1\frac{5}{8}d$, and is coned to rivet shank at an angle of 60°.

Efficiency of Joints. $\eta=\dfrac{p''-d_1}{p''}$. (Following table gives η for steel where $f_t \div f_s=1.2$.)

t.	d.	Single-riv. Lap.	Double-riv. Lap.	Double-riv. Butt.	Treble-riv. Butt.
$\frac{3}{8}$	$\frac{3}{4}$	.57	.73	.84	.93
$\frac{1}{2}$	$\frac{7}{8}$	.54	.70	.82	.92
$\frac{3}{4}$	$1\frac{1}{16}$	.49	.66	.79	.90
1	$1\frac{1}{4}$	.45	.62	.77	.90
$1\frac{1}{4}$	$1\frac{1}{4}$	.40	.57	.73	.87

Riveting in Structural Work (example,—plate girder). Flange area $a=\dfrac{Bm}{hf}$. $\therefore Bm$ (neglecting bending stress on web) $=ahf$ (1). Bm of web $=\dfrac{fth^2}{6}$, or allowing for rivet holes, $=\dfrac{fth^2}{8}$, and Bm (considering bending stress on web) $=hf\left(a+\dfrac{ht}{8}\right)$, and the flange area $a=\dfrac{Bm}{fh}-\dfrac{ht}{8}$ (2).

Riveting: Lower angles to web (in tension), neglecting Moment of Resistance of web to bending; pitch of rivets, $p''=hf_s \div V$, or the vertical shear. Upper angles to web, compression, M. of R. neglected; $p''=hf_s\sqrt{\dfrac{1}{V^2+h^2w^2}}$, where w = total loading per inch of length. p'', h, t in in., a in sq. in., f_s (= least strength of rivet subject to double shear and bearing stress) in lbs. per sq. in., V and w in lbs.

The pitch of rivets joining flange plates to angles is 6 in., excepting at and near the ends of flanges, where $p''=4d$.

Web stiffeners are angles riveted vertically to the web to prevent buckling of the latter. If $t<\dfrac{h}{60}$ the stiffeners should be spaced h in. apart (maximum spacing = 60 in.).

Pins, bolts, and rivets, unless fitting tightly and thoroughly gripping the plates, will be subject to bending stresses and smaller unit stresses must be employed, viz.: for circular sections, $0.75f_s$; for square sections, $0.66f_s$; for square sections, forces acting along diagonal, $0.89f_s$.

Strength of Cotter Joints. d = diam. of rod = breadth of cotter midway between ends $=4\times$ thickness of cotter. Taper of cotter 1 in 30 to 1 in 100. If tapered much greater than 1 in 30, cotters are apt to fly out.

Torsional Stress-Action. External Moment = Moment of Resistance at section, or $wr=f_sS_t$.

Strength of Round Shafts. Moment of Resistance of section $=0.1964f_sd^3$ for solid shafts and $0.1964f_s\left(\dfrac{D^4-d^4}{D}\right)$ for hollow shafts.

Strength of Square Shafts. Moment of Resistance of section $=0.208f_ss^3$, where s = side of square in in.

Factor of Safety for Stiffness = 10 for short shafts; 16 for long shafts.

Strength of Flange Coupling Bolts.

Diam. of bolt $=0.577\sqrt{(\text{diam. of shaft})^3 \div (\text{bolt circle radius} \times \text{No. of bolts})}$.

Strength of Sunk Keys. (Average practice.) Breadth $=\frac{3}{16}$ (diam. of shaft) $+\frac{1}{16}$ in.; Depth $=\frac{1}{8}$ (diam. shaft) $+\frac{1}{8}$ in.; Length $=0.3$ (diam. shaft)$^2 \div$ depth. For splines or keys upon which parts rotating with shaft may also slide axially, interchange the above dimensions for breadth and depth.

The Angle of Torsion, (θ), is the angle through which one end of a shaft turns relatively to the other end under a given stress. (θ = arc $\div$ radius.)

$$\theta=2f_sl \div (d \times \text{Modulus of transverse elasticity, } C).$$

Strength of Helical Springs. For round wire, using shaft equation, $wr=f_s\dfrac{\pi d^3}{16}$, where w = axial pull in lbs., r = radius of coil (to center of wire

section), f_s (safe) = 60,000 (Begtrup and Hartnell). For square wire, $wr = 0.208 f_s s^3$. Deflection $= 2f_s lr \div Cd$, where $l = 2\pi r \times$ No. of turns or spirals, n; d = diam. of wire, and $C = 12{,}000{,}000$. All dimensions in in.

Further, deflection $= 64wnr^3 \div Cd^4$ for round-wire springs, and $= 60.5wnr^3 \div Cs^4$ for square-wire. (Values of f_s and C are for steel wire.)

Conical Springs, round wire, $wr = \dfrac{\pi d^3 f_s}{16}$, where r = largest radius of coil. Deflection $= \dfrac{16wnr^3}{Cd^4}$.

Flat volute (rectangular section of height h, breadth or thickness b),

$$wr = 0.222 b^2 h f_s. \quad \text{Deflection} = \frac{1.8\pi wnr^3(b^2+h^2)}{Cb^3h^3}.$$

Spiral Springs in Torsion.

Round wire, $wr = \pi f_s d^3 \div 32$. Deflection at $r = \dfrac{64wlr^2}{\pi Ed^4}$.

Square wire, $wr = f_s s^3 \div 6$. " " $r = \dfrac{12wlr^2}{Es^4}$.

(l = developed length of spring in inches.)

Bending Stress-Action. In an overhung beam, or cantilever, the upper fibers are in a state of tension and the lower ones in compression, while in a supported beam, or girder, the opposite is the case. There exists therefore an intermediate longitudinal section where these stresses are zero in value. The intersection of this longitudinal section and a vertical cross-section is a line called the Neutral Axis, which passes through the center of figure (or gravity) of the cross-section. Consider two small areas, a_t and a_c (distant y_t and y_c from neutral axis), and let ρ be the radius of curvature of the neutral longitudinal section of the beam when under bending stress. Then, assuming the beam or bar to be bent into a circular ring, l of bar (before bending) $= 2\pi\rho$; l (after bending), or circumference of bar at area $a_t = 2\pi(\rho + y_t)$, in tension, and l at area $a_c = 2\pi(\rho - y_c)$, in compression. Consequently, the strain on fibers at $a_t = 2\pi(\rho + y_t) - 2\pi\rho = 2\pi y_t$, and strain at $a_c = 2\pi\rho - 2\pi(\rho - y_c) = 2\pi y_c$; but $\Delta = \dfrac{fl}{E}$, generally; $\therefore\ 2\pi y = \dfrac{f \cdot 2\pi\rho}{E}$ and $f = \dfrac{Ey}{\rho}$ (1), and the total stress on a small area a, $= fa = \dfrac{Eya}{\rho}$.

Moment of Resistance. Moment of stress on the small area $a = fay = \dfrac{Eay^2}{\rho}$, and the moment of all stresses on the section $= \dfrac{E}{\rho}\Sigma ay^2$. Σay^2 = Moment of Inertia of the section (or Second Moment) $= I$. $\therefore$ Moment of Resistance $= \dfrac{EI}{\rho}$ (2). Representing the moment in terms of the limiting stress, then, Bending Moment, $B_m = fS$ = Moment of Resistance (3). S is called the Section Modulus (= virtual area $\times$ arm through which it acts). From (1), (2), and (3), $S = \dfrac{I}{y}$, and $B_m = \dfrac{fI}{y}$.

Moments of Inertia of Area.

For Beams.

Section.	$I.$	y (= dist. of furthest fiber from axis.)
Rectangle, axis ‖ to breadth and bisecting section.	$\dfrac{bh^3}{12}$	$\dfrac{h}{2}$
Square, ditto, ($b = h$).	$\dfrac{h^4}{12}$	$\dfrac{h}{2}$
Square, axis bisecting section on diagonal.	$\dfrac{s^4}{12}$	$\dfrac{s\sqrt{2}}{2}$ (s = side of square.)

Hollow rectangle or square, axis as for rectangle above........	$\frac{b_1h_1^3 - bh^3}{12}$	$\frac{h_1}{2}$	
Triangle, axis ‖ to base.........	$\frac{bh^3}{36}$	$\frac{2h}{3}$	
Circle, diameter as axis.........	$\frac{\pi d^4}{64}$	$\frac{d}{2}$	
Hollow circle..................	$\frac{\pi}{64}(d_1^4 - d^4)$	$\frac{d_1}{2}$	(b_1, h_1, and d_1 are outer dimensions.)

For shafts. (Polar Moment of Inertia $= I_p$.)

Section.	I_p.	y.	
Rectangle....................	$\frac{bh(b^2+h^2)}{12}$	$\frac{\sqrt{b^2+h^2}}{2}$	
Square.......................	$\frac{s^4}{6}$	$s \div \sqrt{2} = 0.707s$	
Circle.......................	$\frac{\pi d^4}{32}$	$\frac{d}{2}$	
Hollow circle................	$\frac{\pi(d_1^4 - d^4)}{32}$	$\frac{d_1}{2}$	(d_1 = outer diam.)

The Polar Moment of Inertia $I_p = I + I_1$, where I and I_1 are two Moments of Inertia of the section which are taken at right angles to each other through the c. of g. of the section.

The Radius of Gyration, $r = \sqrt{\frac{I}{\text{area of section}}}$.

Moment of Resistance. Graphic Solution. AB is the neutral axis of the rectangular section $CDHJ$, and CD the line of limiting or greatest stress. The value of any horizontal fiber EF to resist stress is found by projecting the same vertically to the line CD and joining C and D to N. The intercept GM is the value desired. All fibers being thus treated, the sum of the virtual stress areas will be the areas CDN and HJN which each make one force of the couple when multiplied by the limiting stress f. K and L are the centers of gravity of the areas.

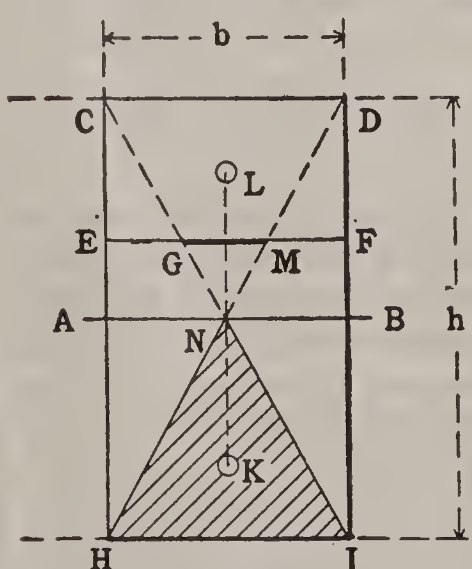


Fig. 2.

Moment of Resistance of rectangular section $= f$ (area CDN or HJN) $\times$ arm $KL = f\left(\frac{bh}{4}\right) \times \frac{2h}{3} = f\frac{bh^2}{6} = fS$.

Moment of Inertia of any Section. Find fS by above method, divide by value of f and multiply by y. ($I = Sy$.) For rectangular section, $S = \frac{bh^2}{6}$, $y = \frac{h}{2}$, and $I = \frac{bh^3}{12}$.

Center of Gravity and Moment of Inertia Determined Graphically (Fig. 3). Beam section 1 2 3 4 5 6 . . . 12. To find center of gravity (considering right half of section): Project each horizontal fiber of section vertically to the arbitrarily assumed line x_1x_1 parallel to base line xx. Join ends of projection to point b and note the intercept on each fiber. The sum of all these fiber intercepts will be the area a 24 17 16 25 26 b a, or A_1. Then, $A_1h = AG$, where A is area of right half of section (sufficient in case of symmetry) and G = distance of center of gravity from xx. Then, $G = A_1h \div A$, which determines the position of neutral axis, zz.

To find I of the section around zz (considering left half of section) Project every horizontal fiber strip of section to ll, the line of limiting stress. join ends of projection to point c (center of gravity) *producing if necessary* until the original strip is crossed, and note the intercepts. The areas 1 a c 14 13 1 (a_1) and c 18 23 22 b c (a_1') are thus found, and on opposite sides of vertical center line. They are the 1st moments. Go through the same process as above with the areas a_1 and a_1', and the second moment areas 1 a c 15 14 1

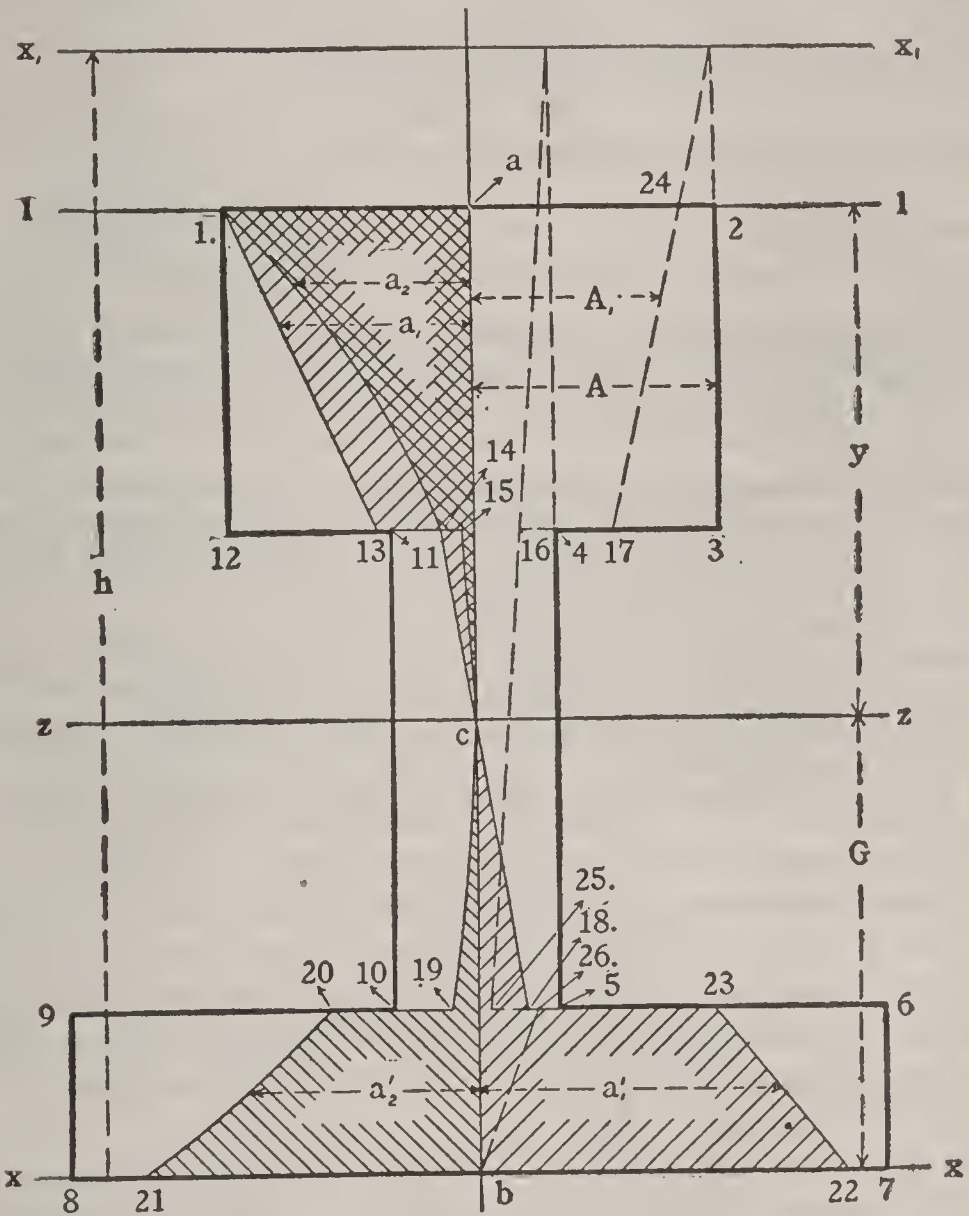


Fig. 3.

(a_2) and c b 21 20 19 c (a_2') will be obtained, both being on the same side of vertical line. Then (doubling the results for the entire section), $I = 2(a_2 + a_2')y^2$ and $S = \frac{I}{y} = 2(a_2 + a_2')y$. In cast-iron beams if $f_c\frac{y_t}{y_c} > f_t$, then f_t is the limiting stress and the line ll should be drawn at a distance y_t from neutral axis.

Position of Center of Gravity. The centers of gravity of regular figures (plane or solid) are the same as their geometrical centers.

Triangle: $\frac{1}{3}$ distance from middle of side to vertex of opposite angle.

Trapezoid: divide into two triangles by a diagonal and join their centers of gravity; repeat process with the other diagonal and the intersection of the lines joining the centers of gravity will be c. of g. of trapezoid.

Sector of circle: on radius bisecting the arc, distance from center $=$ ($2\times$chord$\times$radius)$\div$($3\times$length of arc).

Semicircle: on middle radius, $0.4244r$ from center.

Quadrant: on middle radius, $0.6002r$ from center.

Segment of circle: distance from center $=$ (chord)$^3\div$($12\times$area).

Parabola: $\frac{3}{5}$ length from vertex, and on axis.

Semi-parabola: $\frac{3}{5}$ length from vertex, $\frac{3}{8}$ semi-base from axis.

Cone, Pyramid: in axis, $\frac{1}{4}$ its length from base.

Paraboloid: in axis, $\frac{2}{3}$ its length from vertex.

Frustum of Pyramid: distance from larger base $=\frac{h}{4}\left(\frac{A+3a+2\sqrt{Aa}}{A+a+\sqrt{Aa}}\right)$.

Frustum of Cone: " " " " $=\frac{h}{4}\left(\frac{R^2+r(2R+3r)}{R^2+r(R+r)}\right)$.

h = height; A, a, and R, r = larger and smaller base areas and radii respectively.

Two or more bodies in the same plane: refer to co-ordinate axes. Multiply the weight of each body by the distance from its center of gravity to one of the axes, add the products and divide by the sum of the weights, the result being the distance of the center of gravity of the system from that axis. If bodies are not in a plane, refer them similarly to three rectangular planes.

Moment of Inertia of Compound Shapes. The Moment of Inertia of any section about any axis = the Moment of Inertia about a parallel axis passing through its center of gravity + [area of section $\times$ (distance between axes)^2]. Also, the Radius of Gyration for any section around an axis parallel to another axis through the center of gravity $=$ $\sqrt{(\text{dist. between axes})^2+(\text{radius of gyration around axis through c. of g.})^2}$. By these rules the I and r of "built up" beams and columns may be obtained,—for I, by finding the I of the several components of section about the same axis and adding the results for the combined section.

Bending Moment and Deflection of Beams of Uniform Section. (W = total load on beam.)

I. Beam fixed at one end, concentrated load at the other. Maximum B_m at fixed end $=Wl$. (B_m may be represented by the ordinates of a right-angled triangle having base $=l$ and height $=Wl$.) Deflection $=\frac{Wl^3}{3EI}$.

II. Beam fixed at one end, uniformly distributed load (e.g., wt. of beam). Max. B_m at fixed end $=\frac{Wl}{2}$. (B_m represented by ordinates from base of length l to a semi-parabolic curve having vertex at free end of l and axis perpendicular thereto, and whose semi-parameter $=\frac{l}{W}$). Deflection $=\frac{Wl^3}{8EI}$.

III. Beam, ends supported, concentrated load at center. Max. B_m at center $=\frac{Wl}{4}$. Deflection $=\frac{Wl^3}{48EI}$.

IV. Beam, ends supported, concentrated load at any point. Max. $B_m=\frac{W(l-x)x}{l}$, where x = distance of load from one support. Deflection $=\frac{Wx^2(l-x)^2}{3EIl}$.

V. Beam, ends supported, uniform load. Max. B_m at center $=\frac{Wl}{8}$. Deflection $=\frac{5Wl^3}{384EI}$.

VI. Beam fixed at both ends, centrally loaded. Max. B_m at center and ends $=\frac{Wl}{8}$. Deflection $=\frac{Wl^3}{192EI}$. Points of contra-flexure distant $\frac{l}{4}$ from ends.

VII. Beam fixed at both ends, uniformly loaded. Max B_m at ends $=$

$\frac{Wl}{12}$, $\left(\frac{Wl}{24}\text{, at center}\right)$. Deflection $=\frac{Wl^3}{384EI}$. Points of contra-flexure are $0.211l$ from ends.

VIII. Beam fixed at one end, supported at the other and uniformly loaded. Max. B_m at fixed end $=\frac{Wl}{8}$. Deflection $=\frac{5Wl^3}{926EI}$. Point of contra-flexure $=\frac{l}{4}$ from fixed end.

IX. Beam fixed at one end, supported at the other, and centrally loaded. Max. $B_m=\frac{3Wl}{16}$. Deflection $=\frac{7Wl^3}{768EI}$.

X. Beam loaded at each end with $\frac{W}{2}$, with two supports, each distant x from ends. Max. $B_m=\frac{Wx}{2}$. Deflection, overhang, $=\frac{Wx(3lx-4x^2)}{12EI}$, for middle part, $=\frac{Wx(l-2x)^2}{16EI}$.

XI. Beam, both ends supported, with two symmetrically placed loads $\left(\text{each}=\frac{W}{2}\right)$, each x dist. from support. Max. $B_m=\frac{Wx}{2}$. Deflection $=\frac{Wx(3l^2-4x^2)}{48EI}$.

XII. Beam, fixed at one end, distributed load increasing uniformly from 0 towards fixed end. Max. $B_m=\frac{Wl}{3}$. Deflection $=\frac{Wl^3}{15EI}$.

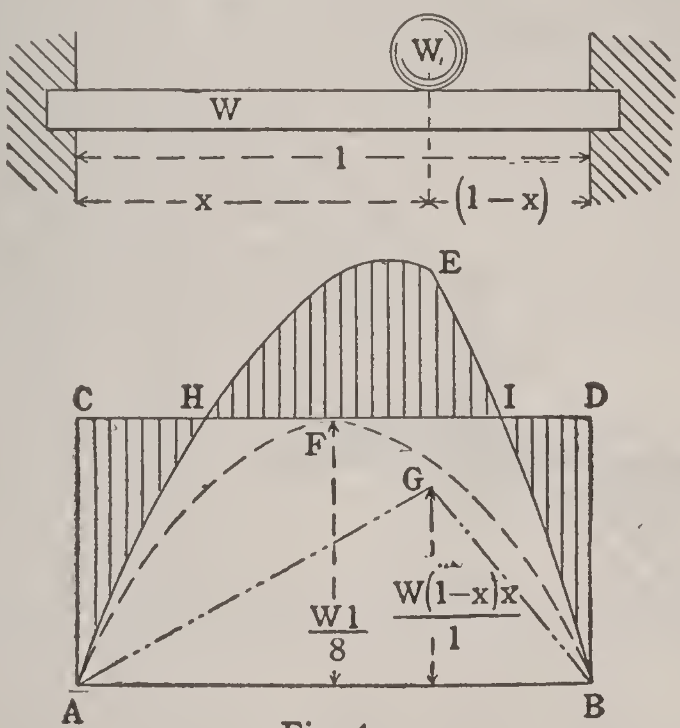


Fig. 4.

XIII. Beam, both ends supported, distributed load increasing uniformly from 0 at center towards ends. Max. $B_m=\frac{Wl}{12}$. Deflection $=\frac{3Wl^3}{320EI}$.

XIV. Same as XIII, but with load increasing uniformly from O at ends to center. Max. $B_m=\frac{Wl}{6}$. Deflection $=\frac{Wl^3}{60EI}$.

XV. Beam overhanging each of two supports by distance x, uniformly distributed load.. $B_m = \frac{Wx^2}{2l}$ at either support, and $\frac{W}{2}(x-0.25l)$ at center. Max. B_m (when $x=0.207l$) $= \frac{3Wl}{140}$.

Combinations of loading may be shown graphically as in Fig. 4. W = uniform load, and W_1 = concentrated load. Consider the beam as merely supported at the ends, with a uniform load (e.g., itself). Then, the parabola AFB, on base AB, and of height $= \frac{Wl}{8}$, is the curve of B_m for W. Again, consider beam as loaded only with W_1. Then, the triangle AGB will be the curve of B_m for W_1, and, by adding the ordinates of these curves a new curve $AHEIB$ is obtained, which is the curve of B_m for the combined loads on a freely supported beam. Again, consider the beam as fixed. The B_m of the supported beam is now opposed by the reaction of the wall, which is a constant strain and whose B_m curve is the rectangle $ACDB$, equal in area to $AHEIB$. The algebraic sum of these bending moments gives for the fixed beam the shaded B_m curve $ACHEIDBIHA$, and the intersections at H and I determine the points of contra-flexure. The portions CH and ID are strained as cantilevers, the upper sides being in tension, while the part HI is strained as a supported girder, with tension on lower side.

The B_m curve for a moving load (e.g., that on a travelling-crane girder) is parabolic, with a maximum at center equal to $\frac{Wl}{4}$.

Shear Stresses. **The vertical shear stress** caused by a concentrated load is represented by the ordinates of a rectangular area having a length = dist. from point of support to point of max. B_m, and a height = reaction at point of support. The vert. shear stress caused by a uniformly distributed load is represented by the ordinates of a right-angled triangular area having base as above, and height at point of support = reaction at that point. Thus, in Fig. 5, rectangles 1 2 3 4 and 2 5 6 7 are for concentrated load W_1 (see Fig. 4), and triangles 1 8 9 and 9 10 7 for distributed load W. The algebraic sum of these areas gives areas 1 11 12 and 12 13 14 15 7 12, any ordinate of which shows the vertical shear stress of the combined loads at the point where ordinate is erected. Heights 1 4, 6 7 and 1 11, 7 15 represent the reactions or proportions of W_1 and W respectively sustained by the points of support.

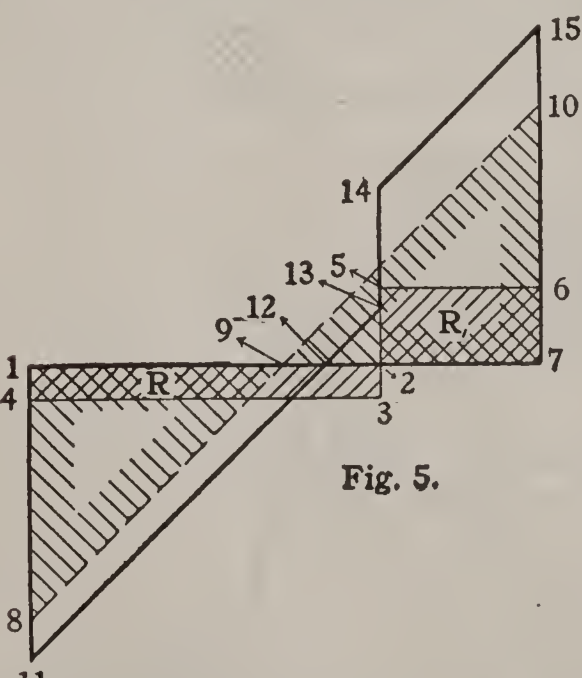


Fig. 5.

Horizontal shear stress. If a summation of the horizontal forces (tensile and compressive) is taken, proceeding from the upper or lower fibre to the neutral axis, it will be found that the max. hor. shear stress is at the neutral axis, and, in a rectangular beam, at any section: Max. hor. shear stress = (3 × Vert. shear at the section considered) ÷ $2bd$, where b and d are breadth and depth of beam. In long beams the shear is small compared with the bending stress and is fully taken care of by the surplus section; in short beams it should be considered.

Continuous Beams. (Reactions on supports in terms of W_1, the uniform load on each span.)

3 supports	3	10	3	....	....	....	...	...	...	...	each $\times W_1 \div$ 8
4 "	4	11	11	4	....	....	...	...	...	...	" " $\div$ 10
5 "	11	32	26	32	11	....	...	...	...	...	" " $\div$ 28
6 "	15	43	37	37	43	15	...	...	...	...	" " $\div$ 38
7 "	41	118	108	106	108	118	41	...	...	...	" " $\div$ 104
8 "	56	161	137	143	143	137	161	56	...	...	" " $\div$ 142
9 "	152	440	374	392	386	392	374	440	152	...	" " $\div$ 388
10 "	209	601	511	535	529	529	535	511	601	209	" " $\div$ 530

The Allowable Deflection for cantilevers is $\frac{1}{80}$ in. per foot of span, and $\frac{1}{40}$ in. per ft. of span for girders.

Beams of Uniform Strength (Rectangular Section).—With constant breadth, the depth varies as the ordinates of: I, a semi-parabola with vertex at loaded end; II, a triangle, base at fixed end; III and IV, two semi-parabolas, vertices at supports, bases joining at load point; V, a semi-ellipse. With constant depth the breadth varies as the ordinates of: I, a triangle, base at fixed end; II, distance between two convex parabolas whose vertices touch at free end; III and IV, two triangles, bases at load point: V, distance between two symmetrical concave parabolas intersecting at points of support. (I, II, III, etc., refer to conditions of loading under the heading of Bending Moment and Deflection of Beams, *ante*.)

Strength of Circular Flat Plates of Radius r (Grashof).—Plate supported at circumference and uniformly loaded: $f=0.833pr^2\div t^2$. Same loading, plate fixed at circumference: $f=0.666pr^2\div t^2$. Plate supported at circumference, loaded centrally with w (of radius r_1): $f=\left(1.333\log\frac{r}{r_1}+1\right)\frac{w}{\pi t^2}$.

Strength of Square and Rectangular Flat Plates, Uniformly Loaded (Unwin).—Rectangular plate, fixed at edges: $f=0.5b^2l^4p\div(b^4+l^4)t^2$, where b=breadth and l=length of plate in in. Square plate, fixed at edges: $f=0.25ps^2\div t^2$, where s=side in in. Surface supported by stays: $f=0.222ps^2\div t^2$, where s=distance in in. between the centers of stays, which are arranged in rows. f=working stress in lbs.

Strength of Flat Stayed Surfaces. (See Steam Boilers.)

Strength of Laminated Steel Springs. $w=\frac{fnbt^2}{6l}$. Deflection, $\Delta=\frac{fl^2}{Et}$, where w=max. static load on one end of a semi-elliptic, or $\frac{1}{2}$ max. load on full elliptic spring; f=allowable stress in lbs. per sq. in. (varying according to homogeneity and temper)=90,000 for $\frac{1}{4}$-in. plates, 80,000 for $\frac{3}{8}$-in., and 75,000 for $\frac{1}{2}$-in.; n=no. of plates; l=half span in ins.; E=30,000,000. (Reuleaux and Gaines.)

Combined Stresses.

Bending and Tension (Load parallel to axis at distance r).—Bending action $=wr=f_rS=f_tcS$; tensile action $=w=f_ta$. Combined max. tensile stress on edge nearest axis of $w=f_t'=w\left(\frac{1}{a}+\frac{r}{cS}\right)$. (See Modulus of Rupture.)

Strength of Crane Hooks. $w=abf_t\div C$, where a=radius of inside of hook or sling, h=breadth of hook on hor. section through center of inside hook circle, b=thickness of section, w=load in lbs., f_t safe=13,000 to 17,000 lbs.

Values of C:

	$h\div a=$ 1	1.5	2	2.5	3	4
Rectangular section,	C=12.6	7.25	5.07	3.92	3.22	2.41
Trapezoidal section,	C=15	8.96	6.42	5.06	4.18	3.28
Elliptical "	C=21.5	12.58	8.89	6.92	5.73	

Distance from center of hook circle to shoulder on bolt end $=2h$. Diam. of bolt end $d_1=\sqrt{\frac{w}{4,267}}$. In trapezoidal sections, the wide edge b should be next to rope or chain; narrow edge $b_1=b\div\left(\frac{h}{a}+1\right)$.

(Ing. Taschenbuch).

Towne gives the following proportions: Neck $=d$ (taken as unit); turned shank $=0.87d$; sling diam. $=1.65d$; diam. of tip on hor. diam. of sling $=0.7d$; radial width of flattened wedge section on hor. sling diam. $=1.4d$; thickness of inner wedge edge $=0.875d$; do., outer edge $=0.25d$; width at mouth of sling $=1.25d$. Safe dead load in lbs. $=1{,}500d^2$, where d is in inches.

Reuleaux gives the following: $2a=1.95d_1=0.039\sqrt{w}=h=1.5b=2\times$ diam. of hook tip on hor. line through c. of hook sling, $=1.33\times$ width at hook opening. These values agree fairly well with the Taschenbuch formulas (taking $f_t=13{,}000$). (Compare with formula $f_t=w\left(\frac{1}{a}+\frac{r}{S}\right)$.)

Bending and Compression. Substitute f_c for f_t in formulas for bending and tension. Example: ship's davits.

Columns and Struts. While these are cases involving bending and compression, their action is more complex. Where $l<12d$ they are calculated for direct crushing only; longer columns bend before breaking.

Gordon's Formulas. f breaking $=\dfrac{a}{1+b\frac{l^2}{r^2}}$, both ends fixed or flat;

$=\dfrac{a}{1+1.8b\left(\frac{l^2}{r^2}\right)}$, one end fixed, other hinged or round;

$=\dfrac{a}{1+4b\left(\frac{l^2}{r^2}\right)}$, both ends hinged or round;

where l=length in ins., r=*least* radius of gyration, and a and b are as follows:

	a.	b.	
Cast Iron..............	80,000	$\frac{1}{6{,}400}$	
W. I. and very soft steel..	36,000–40,000	$\frac{1}{36{,}000}$ to $\frac{1}{40{,}000}$	
Medium Steel..........	67,000	$\frac{1}{22{,}400}$	
Hard Steel.............	114,000	$\frac{1}{14{,}400}$	
Dry Timber............	7,200	$\frac{1}{3{,}000}$	
Soft Steel..............	15,000	$\frac{1}{13{,}500}$	Am. Bridge Co. Practice. Safe values.
Medium Steel..........	17,000	$\frac{1}{11{,}000}$	

Then, w (lbs.) $=\dfrac{f\text{ (breaking) in lbs. per sq. in.}\times\text{area of section in sq. in.}}{\text{Factor of safety.}}$

For W. I. and steel, factor of safety $=4$ for dead load, and 5 for moving load. For C. I. not less than 8.

Prof. Lanza states as the result of experiments that Gordon's formulas do not apply in the case of cast-iron columns, and he recommends 5,000 lbs. per sq. in. as the highest allowable safe loading, the length of column not to exceed 20 times its diameter and the metal to be of thickness sufficient to insure sound castings.

Eccentric Loading. When the resultant of the load does not pass through the c. of g. of the section, let r=distance between resultant and c. of g. of section; I its moment of inertia about an axis in its plane passing through the c. of g. and perpendicular to r; y=distance between said axis and fibre under greatest compression; w=total pressure on section. Then $f=\frac{w}{a}+\frac{wry}{I}$. Assume a section, compute f, and if it exceeds safe value (5,000 for C. I.) assume another section and compute f until a safe

value is found. Eccentric loading in buildings is due to the unequal distribution of loads on floors. If liable to occur only in rare cases, f may be taken at 8,000 lb. per sq. in. for C. I.

Safe Loads for Round and Square Cast Iron Columns. (City Building Laws, 1897.) Safe load in tons of 2,000 lb. $= \dfrac{Ca}{1+\dfrac{l^2}{bd^2}}$.

	New York.	Boston.	Chicago.
C (round or sq.)........... =	8	5	5
b (square)................ =	500	1,067	800
b (round)................. =	400	800	600

Resistance of Hollow Cylinders to Collapse. (See Furnace Flues under "Steam Boilers.")

Torsion and Bending. This combination of stresses exists to a greater or less extent in all shafting. Equivalent twisting moment $=2\times$ equivalent bending moment $=B_m+\sqrt{B_m^2+T_m^2}$, where T_m = twisting moment $= fsI_p \div y$.

Torsion and Compression. (Propeller shaft.) $w=\dfrac{\pi^2 EI}{l^2}-\dfrac{T_m^2}{4EI}$. A safety factor of 5 should be used.

Modulus of Rupture. The ultimate stress obtained from the momental formula in breaking a solid beam by bending will usually be found much greater than f_t breaking. Modulus of Rupture $f_r=cf_t$, where c generally $=2$ for circular and square (one diagonal vertical) sections, 1.5 for square and rectangular sections, and unity for I and T sections. The values of c depend however on the material: Rectangular sections; Fir, 0.52 to 0.94; Oak, 0.7 to 1; Pitch-pine, 0.8 to 2.2; C. I., 2; W. I., 1.6; Forged steel, 1.47; Gun metal, 1. Circular sections: C. I., 2.35; W. I., 1.75; Forged steel, 1.6; Gun metal, 1.9. I sections: C. I., $1+$(web thickness $\div$ flange width).

CARNEGIE ROLLED STRUCTURAL STEEL.

In the following tables, w = weight in lbs. per lineal foot, a = area of section in sq. in., h = depth of beam or channel in in., b = width of flange in in., t = thickness of web in in.

x, x_1, x_2 = distance between c. of g. of section and (1) outside of channel web; (2) outside of flange on T; (3) back of flange of equal leg angle.

I, r, S = Moment of inertia, radius of gyration and section modulus, where
Neutral axis is perpendicular to web at center (Beams and channels).
" " " parallel to longer flange (Unequal leg angles).
" " " through c. of g. parallel to flange (Ts and equal leg angles).
" " " through c. of g. perpendicular to web (Zs).

I', r' = Moment of inertia and radius of gyration, where
Neutral axis is coincident with center line of web (Beams).
" " " parallel to center line of web (Channels).
" " " " " shorter flange (Unequal leg angles).
" " " through c. of g. coincident with stem (Ts).
" " " " " " " " " web (Zs).

r'' = Least radius of gyration, neutral axis diagonal.

S' = Section modulus, where
Neutral axis is through c. of g. coincident with stem (Ts).
" " " " " " " " " web (Zs).
" " " parallel to shorter flange (Unequal leg angles).

C = Coefficient of strength for fibre stress of 16,000 lbs. per sq. in. for beams, channels, and Zs, and 12,000 lbs. per sq. in. for Ts.

$C=WL=8M=\dfrac{8fS}{12}$, where f = 12,000 to 16,000 lbs.; M = moment of forces in ft.-lbs., W = safe uniformly distributed load in lbs., L = span in feet. For concentrated load at middle of span use one-half the value of C in the tables. For quiescent loads f = 16,000 lbs. per sq. in.; for moving loads, 12,500 lbs., and, if impact is considerable, f = 8,000 lbs.

For columns or struts consisting of two latticed channels, r of column section (neut. axis in center of section ∥ to webs) = distance between c. of g.

of channel and center of column section (neglecting the *I*s of channels around their own axes,—a slight error on the safe side).

Carnegie Steel I Beams.

(Sizes with * prefixed are standard, others are special.)

h.	*w.*	*a.*	*t.*	*b.*	*I.*	*I′.*	*r.*	*r′.*	*S.*	*C.*
24 in.	100	29.41	0.75	7.25	2380.3	48.56	9	1.28	198.4	2115800
	95	27.94	.69	7.19	2309.6	47.1	9.09	1.3	192.5	2052900
	90	26.47	.63	7.13	2239.1	45.7	9.2	1.31	186.6	1990300
	85	25	.57	7.07	2168.6	44.35	9.31	1.33	180.7	1927600
*	80	23.32	.50	7	2087.9	42.86	9.46	1.36	174	1855900
20	100	29.41	.88	7.28	1655.8	52.65	7.5	1.34	165.6	1766100
	95	27.94	.81	7.21	1606.8	50.78	7.58	1.35	160.7	1713900
	90	26.47	.74	7.14	1557.8	48.98	7.67	1.36	155.8	1661600
	85	25	.66	7.06	1508.7	47.25	7.77	1.37	150.9	1609300
*	80	23.73	.60	7	1466.5	45.81	7.86	1.39	146.7	1564300
	75	22.06	.65	6.40	1268.9	30.25	7.58	1.17	126.9	1353500
	70	20.59	.58	6.32	1219.9	29.04	7.7	1.19	122	1301200
*	65	19.08	.50	6.25	1169.6	27.86	7.83	1.21	117	1247600
18	70	20.59	.72	6.26	921.3	24.62	6.69	1.09	102.4	1091900
	65	19.12	.64	6.18	881.5	23.47	6.79	1.11	97.9	1044800
	60	17.65	.56	6.09	841.8	22.38	6.91	1.13	93.5	997700
*	55	15.93	.46	6	795.6	21.19	7.07	1.15	88.4	943000
15	100	29.41	1.18	6.77	900.5	50.98	5.53	1.31	120.1	1280700
	95	27.94	1.09	6.68	872.9	48.37	5.59	1.32	116.4	1241500
	90	26.47	.99	6.58	845.4	45.91	5.65	1.32	112.7	1202300
	85	25	.89	6.48	817.8	43.57	5.72	1.32	109	1163000
*	80	23.81	.81	6.4	795.5	41.76	5.78	1.32	106.1	1131300
	75	22.06	.88	6.29	691.2	30.68	5.60	1.18	92.2	983000
	70	20.59	.78	6.19	663.6	29	5.68	1.19	88.5	943800
	65	19.12	.69	6.1	636	27.42	5.77	1.2	84.8	904600
*	60	17.67	.59	6	609	25.96	5.87	1.21	81.2	866100
	55	16.18	.66	5.75	511	17.06	5.62	1.02	68.1	726800
	50	14.71	.56	5.65	483.4	16.04	5.73	1.04	64.5	687500
	45	13.24	.46	5.55	455.8	15.00	5.87	1.07	60.8	648200
*	42	12.48	.41	5.5	441.7	14.62	5.95	1.08	58.9	628300
12	55	16.18	.82	5.61	321	17.46	4.45	1.04	53.5	570600
	50	14.71	.70	5.49	303.3	16.12	4.54	1.05	50.6	539200
	45	13.24	.58	5.37	285.7	14.89	4.65	1.06	47.6	507900
*	40	11.84	.46	5.25	268.9	13.81	4.77	1.08	44.8	478100
	35	10.29	.44	5.09	228.3	10.07	4.71	.99	38	405800
*	31.5	9.26	.35	5	215.8	9.50	4.83	1.01	36	383700
10	40	11.76	.75	5.10	158.7	9.50	3.67	.90	31.7	338500
	35	10.29	.60	4.95	146.4	8.52	3.77	.91	29.3	312400
	30	8.82	.46	4.8	134.2	7.65	3.9	.93	26.8	286300
*	25	7.37	.31	4.66	122.1	6.89	4.07	.97	24.4	260500
9	35	10.29	.73	4.77	111.8	7.31	3.29	.84	24.8	265000
	30	8.82	.57	4.61	101.9	6.42	3.4	.85	22.6	241500
	25	7.35	.41	4.45	91.9	5.65	3.54	.88	20.4	217900
*	21	6.31	.29	4.33	84.9	5.16	3.67	.90	18.9	201300
8	25.5	7.50	.54	4.27	68.4	4.75	3.02	.80	17.1	182500
	23	6.76	.45	4.18	64.5	4.39	3.09	.81	16.1	172000
	20.5	6.03	.36	4.09	60.6	4.07	3.17	.82	15.1	161600
*	18	5.33	.27	4	56.9	3.78	3.27	.84	14.2	151700
7	20	5.88	.46	3.87	42.2	3.24	2.68	.74	12.1	128600
	17.5	5.15	.35	3.76	39.2	2.94	2.76	.76	11.2	119400
*	15	4.42	.25	3.66	36.2	2.67	2.86	.78	10.4	110400
6	17¼	5.07	.48	3.58	26.2	2.36	2.27	.68	8.7	93100
	14¾	4.34	.35	3.45	24	2.09	2.35	.69	8	85300
*	12¼	3.61	.23	3.33	21.8	1.85	2.46	.72	7.3	77500
5	14¾	4.34	.50	3.29	15.2	1.7	1.87	.63	6.1	64600
	12¼	3.60	.36	3.15	13.6	1.45	1.94	.63	5.4	58100
*	9¾	2.87	.21	3	12.1	1.23	2.05	.65	4.8	51600

Carnegie Steel I Beams.—*Continued.*

h.	*w.*	*a.*	*t.*	*b.*	*I.*	*I′.*	*r.*	*r′.*	*S.*	*C.*
4 in.	10.5	3.09	.41	2.88	7.1	1.01	1.52	.57	3.6	38100
	9.5	2.79	.34	2.8	6.7	0.93	1.55	.58	3.4	36000
	8.5	2.5	.26	2.73	6.4	.85	1.59	.58	3.2	33900
*	7.5	2.21	.19	2.66	6	.77	1.64	.59	3	31800
3	7.5	2.21	.36	2.52	2.9	.60	1.15	.52	1.9	20700
	6.5	1.91	.26	2.42	2.7	.53	1.19	.52	1.8	19100
*	5.5	1.63	.17	2.33	2.5	.46	1.23	.53	1.7	17600

Carnegie Steel Channels.

(Sizes with * prefixed are standard, others are special.)

h.	*w.*	*a.*	*t.*	*b.*	*I.*	*I′.*	*r.*	*r′.*	*S.*	*C.*	*x.*
15 in.	55	16.18	0.82	3.82	430.2	12.19	5.16	0.868	57.4	611900	0.823
	50	14.71	.72	3.72	402.7	11.22	5.23	.873	53.7	572700	.803
	45	13.24	.62	3.62	375.1	10.29	5.32	.882	50	533500	.788
	40	11.76	.52	3.52	347.5	9.39	5.43	.893	46.3	494200	.783
	35	10.29	.43	3.43	320	8.48	5.58	.908	42.7	455000	.789
*	33	9.9	.40	3.40	312.6	8.23	5.62	.912	41.7	444500	.794
12	40	11.76	.76	3.42	197	6.63	4.09	.751	32.8	350200	.722
	35	10.29	.64	3.3	179.3	5.9	4.17	.757	29.9	318800	.694
	30	8.82	.51	3.17	161.7	5.21	4.28	.768	26.9	287400	.677
	25	7.35	.39	3.05	144	4.53	4.43	.785	24	256100	.678
*	20.5	6.03	.28	2.94	128.1	3.91	4.61	.805	21.4	227800	.704
10	35	10.29	.82	3.18	115.5	4.66	3.35	.672	23.1	246400	.695
	30	8.82	.68	3.04	103.2	3.90	3.42	.672	20.6	220300	.651
	25	7.35	.53	2.89	91	3.40	3.52	.680	18.2	194100	.62
	20	5.88	.38	2.74	78.7	2.85	3.66	.696	15.7	168000	.609
*	15	4.46	.24	2.6	66.9	2.30	3.87	.718	13.4	142700	.639
9	25	7.35	.62	2.82	70.7	2.98	3.10	.637	15.7	167600	.615
	20	5.88	.45	2.65	60.8	2.45	3.21	.646	13.5	144100	.585
	15	4.41	.29	2.49	50.9	1.95	3.40	.665	11.3	120500	.59
*	13¼	3.89	.23	2.43	47.3	1.77	3.49	.674	10.5	112200	.607
8	21¼	6.25	.58	2.62	47.8	2.25	2.77	.6	11.9	127400	.587
	18¾	5.51	.49	2.53	43.8	2.01	2.82	.603	11	116900	.567
	16¼	4.78	.40	2.44	39.9	1.78	2.89	.610	10	106400	.556
	13¾	4.04	.31	2.35	36	1.55	2.98	.619	9	96000	.557
*	11¼	3.35	.22	2.26	32.3	1.33	3.11	.63	8.1	86100	.576
7	19¾	5.81	.63	2.51	33.2	1.85	2.39	.565	9.5	101100	.583
	17¼	5.07	.53	2.41	30.2	1.62	2.44	.564	8.6	92000	.555
	14¾	4.34	.42	2.3	27.2	1.40	2.50	.568	7.8	82800	.535
	12¼	3.60	.32	2.2	24.2	1.19	2.59	.575	6.9	73700	.528
*	9¾	2.85	.21	2.09	21.1	.98	2.72	.586	6	66800	.546
6	15.5	4.56	.56	2.28	19.5	1.28	2.07	.529	6.5	69500	.546
	13	3.82	.44	2.16	17.3	1.07	2.13	.529	5.8	61600	.517
	10.5	3.09	.32	2.04	15.1	.88	2.21	.534	5	53800	.503
*	8	2.38	.20	1.92	13	.70	2.34	.542	4.3	46200	.517
5	11.5	3.38	.48	2.04	10.4	.82	1.75	.493	4.2	44400	.508
	9	2.65	.33	1.89	8.9	.64	1.83	.493	3.5	37900	.481
*	6.5	1.95	.19	1.75	7.4	.48	1.95	.498	3	31600	.489
4	7¼	2.13	.33	1.73	4.6	.44	1.46	.455	2.3	24400	.463
	6¼	1.84	.25	1.65	4.2	.38	1.51	.454	2.1	22300	.458
*	5¼	1.55	.18	1.58	3.8	.32	1.56	.453	1.9	20200	.464
3	6	1.76	.36	1.6	2.1	.31	1.08	.421	1.4	14700	.459
	5	1.47	.26	1.5	1.8	.25	1.12	.415	1.2	13100	.443
*	4	1.19	.17	1.41	1.6	.20	1.17	.409	1.1	11600	.443

Carnegie T Shapes (Selected).

Flange × Stem, ins.	$w.$	$a.$	$x_1.$	$I_\circ$	$S.$	$r.$	$I'.$	$S'.$	$r'.$	$C.$
4×5	15.7	4.56	1.56	10.7	3.10	1.54	2.8	1.41	0.79	24800
4×5	12.3	3.54	1.51	8.5	2.43	1.56	2.1	1.06	.78	19410
4×4½	14.8	4.29	1.37	8	2.55	1.37	2.8	1.41	.81	20400
4×4	13.9	4.02	1.18	5.7	2.02	1.2	2.8	1.4	.84	16170
3×4	10.6	3.12	1.32	4.8	1.78	1.25	1.09	.72	.60	14270
3×4	9.3	2.73	1.29	4.3	1.57	1.26	.93	.62	.59	12540
3×3½	9.8	2.88	1.11	3.3	1.37	1.08	1.31	.88	.68	10990
3×3½	8.6	2.49	1.09	2.9	1.21	1.09	.93	.62	.61	9680
3×3	9	2.67	.92	2.1	1.01	.9	1.08	.72	.64	8110
2½×3	6.2	1.8	.92	1.6	.76	.94	.44	.35	.51	6110
2½×2¾	5.9	1.71	.83	1.2	.6	.83	.44	.35	.51	4830
2½×2½	5.6	1.62	.74	.87	.5	.74	.44	.35	.52	4000
2¼×2¼	5	1.44	.69	.66	.42	.68	.33	.30	.48	3360
2¼×2¼	4.2	1.2	.66	.51	.32	.67	.25	.22	.47	2600
2×2	3.7	1.08	.59	.36	.25	.6	.18	.18	.42	2000
1¾×1¾	3.2	.9	.54	.23	.19	.51	.12	.14	.37	1540
1½×1½	2.6	.75	.42	.15	.14	.49	.08	.10	.34	1150
1½×1½	2	.54	.44	.11	.11	.45	.06	.07	.31	860
1¼×1¼	2.1	.60	.40	.08	.10	.36	.05	.07	.27	760
1×1	1.23	.36	.32	.03	.05	.29	.02	.04	.21	370
1×1	0.87	.26	.29	.02	.03	.29	.01	.02	.21	270

Carnegie Steel Angles with Equal Legs.

Max. and Min. Wts. Special Sizes marked *.

Size.	$t.$	$w.$	$a.$	$x_2.$	$I_\circ$	$S.$	$r.$	$r''.$
8×8	1⅛	56.9	16.73	2.41	97.97	17.53	2.42	1.55
8×8	½	26.4	7.75	2.19	48.63	8.37	2.5	1.58
6×6	1	37.4	11	1.86	35.46	8.57	1.8	1.16
6×6	⅜	14.9	4.36	1.64	15.39	3.53	1.88	1.19
*5×5	1	30.6	9	1.61	19.64	5.8	1.48	0.96
*5×5	⅜	12.3	3.61	1.39	8.74	2.42	1.56	.99
4×4	13/16	19.9	5.84	1.29	8.14	3.01	1.18	.77
4×4	5/16	8.2	2.4	1.12	3.71	1.29	1.24	.79
3½×3½	13/16	17.1	5.03	1.17	5.25	2.25	1.02	.67
3½×3½	5/16	7.2	2.09	.99	2.45	.98	1.08	.69
3×3	⅝	11.5	3.36	.98	2.62	1.3	.88	.57
3×3	¼	4.9	1.44	.84	1.24	.58	.93	.59
*2¾×2¾	½	8.5	2.5	.87	1.67	.89	.82	.52
*2¾×2¾	¼	4.5	1.31	.78	.93	.48	.85	.55
2½×2½	½	7.7	2.25	.81	1.23	.73	.74	.47
2½×2½	3/16	3.1	.9	.69	.55	.30	.78	.49
*2¼×2¼	½	6.8	2	.74	.87	.58	.66	.43
*2¼×2¼	3/16	2.8	.81	.63	.39	.24	.70	.44
2×2	7/16	5.3	1.56	.66	.54	.40	.59	.39
2×2	3/16	2.5	.72	.57	.28	.19	.62	.40
1¾×1¾	7/16	4.6	1.3	.59	.35	.30	.51	.33
1¾×1¾	3/16	2.2	.62	.51	.18	.14	.54	.35
1½×1½	⅜	3.4	.99	.51	.19	.19	.44	.29
1½×1½	⅛	1.3	.36	.42	.08	.07	.46	.30
1¼×1¼	5/16	2.4	.69	.42	.09	.109	.36	.23
1¼×1¼	⅛	1.1	.3	.35	.044	.049	.38	.25
1×1	¼	1.5	.44	.34	.037	.056	.29	.19
1×1	⅛	.8	.24	.3	.022	.031	.31	.20
*⅞×⅞	3/16	1.0	.29	.29	.019	.033	.26	.18
*⅞×⅞	⅛	.7	.21	.26	.014	.023	.26	.19
¾×¾	3/16	.9	.25	.26	.012	.024	.22	.16
¾×¾	⅛	.6	.17	.23	.009	.017	.23	.17

Carnegie Steel Angles with Unequal Legs.

Max. and Min. Wts. Special Sizes marked *.

Size.	t.	w.	a.	I.	I'.	S.	S'.	r.	r'.	r''
*8×3½	$\frac{11}{20}$	20.5	6.02	4.92	39.96	1.79	7.99	0.9	2.58	0.74
*7×3½	1	32.3	9.5	7.53	45.37	2.96	10.58	.89	2.19	.88
*7×3½	$\frac{7}{16}$	15	4.4	3.95	22.56	1.47	5.01	.95	2.26	.89
6×4	1	30.6	9	10.75	30.75	3.79	8.02	1.09	1.85	.85
6×4	$\frac{3}{8}$	12.3	3.61	4.9	13.47	1.6	3.32	1.17	1.93	.88
6×3½	1	28.9	8.5	7.21	29.24	2.9	7.83	.92	1.85	.74
6×3½	$\frac{3}{8}$	11.7	3.42	3.34	12.86	1.23	3.25	.99	1.94	.77
*5×4	$\frac{7}{8}$	24.2	7.11	9.23	16.42	3.31	4.99	1.14	1.52	.84
*5×4	$\frac{3}{8}$	11	3.23	4.67	8.14	1.57	2.34	1.2	1.59	.86
5×3½	$\frac{7}{8}$	22.7	6.67	6.21	15.67	2.52	4.88	.96	1.53	.75
5×3½	$\frac{5}{16}$	8.7	2.56	2.72	6.6	1.02	1.94	1.03	1.61	.76
5×3	$\frac{13}{16}$	19.9	5.84	3.71	13.98	1.74	4.45	.80	1.55	.64
5×3	$\frac{5}{16}$	8.2	2.4	1.75	6.26	.75	1.89	.85	1.61	.66
*4½×3	$\frac{13}{16}$	18.5	5.43	3.6	10.33	1.71	3.62	.81	1.38	.64
*4½×3	$\frac{5}{16}$	7.7	2.25	1.73	4.69	.76	1.54	.88	1.44	.66
*4×3½	$\frac{13}{16}$	18.5	5.43	5.49	7.77	2.30	2.92	1.01	1.19	.72
*4×3½	$\frac{5}{16}$	7.7	2.25	2.59	3.56	1.01	1.26	1.07	1.26	.73
4×3	$\frac{13}{16}$	17.1	5.03	3.47	7.34	1.68	2.87	.83	1.21	.64
4×3	$\frac{5}{16}$	7.2	2.09	1.65	3.38	.74	1.23	.89	1.27	.65
3½×3	$\frac{13}{16}$	15.8	4.62	3.33	4.98	1.65	2.20	.85	1.04	.62
3½×3	$\frac{5}{16}$	6.6	1.93	1.58	2.33	.72	.96	.90	1.1	.63
3½×2½	$\frac{11}{16}$	12.5	3.65	1.72	4.13	.99	1.85	.67	1.06	.53
3½×2½	$\frac{1}{4}$	4.9	1.44	.78	1.80	.41	.75	.74	1.12	.54
*3¼×2	$\frac{9}{16}$	9	2.64	.75	2.64	.53	1.30	.53	1	.44
*3¼×2	$\frac{1}{4}$	4.3	1.25	.4	1.36	.26	.63	.57	1.04	.45
3×2½	$\frac{9}{16}$	9.5	2.78	1.42	2.28	.82	1.15	.72	.91	.52
3×2½	$\frac{1}{4}$	4.5	1.31	.74	1.17	.40	.56	.75	.95	.53
*3×2	$\frac{1}{2}$	7.7	2.25	.67	1.92	.47	1.00	.55	.92	.43
*3×2	$\frac{1}{4}$	4.1	1.19	.39	1.09	.25	.54	.57	.95	.43
2½×2	$\frac{1}{2}$	6.8	2	.64	1.14	.46	.70	.56	.75	.42
2½×2	$\frac{3}{16}$	2.8	.81	.29	.51	.20	.29	.60	.79	.43
*2¼×1½	$\frac{1}{2}$	5.6	1.63	.26	.75	.26	.54	.40	.68	.39
*2¼×1½	$\frac{3}{16}$	2.3	.67	.12	.34	.11	.23	.43	.72	.40
*2×1⅜	$\frac{1}{4}$	2.7	.78	.12	.37	.12	.23	.39	.63	.30
*2×1⅜	$\frac{3}{16}$	2.1	.60	.09	.24	.09	.18	.40	.63	.31
*1⅜×1	$\frac{1}{4}$	1.9	.53	.04	.09	.05	.09	.27	.41	.22
*1⅜×1	$\frac{1}{8}$	1	.28	.02	.05	.03	.06	.29	.44	.22

Carnegie Steel Z Bars.

(Dimensions: thickness × width of flange × depth of web.)

Dimensions.	w.	a.	I.	I'	S.	S'.	r.	r'.	r''	C.
$\frac{3}{8}$×3½×6	15.6	4.59	25.32	9.11	8.44	2.75	2.35	1.41	0.83	90000
$\frac{7}{16}$×3$\frac{9}{16}$×6$\frac{1}{16}$	18.3	5.39	29.8	10.95	9.83	3.27	2.35	1.43	.84	104800
$\frac{1}{2}$×3⅝×6⅛	21	6.19	34.36	12.87	11.22	3.81	2.36	1.44	.84	119700
$\frac{9}{16}$×3½×6	22.7	6.68	34.64	12.59	11.52	3.91	2.28	1.37	.81	123200
$\frac{5}{8}$×3$\frac{9}{16}$×6$\frac{1}{16}$	25.4	7.46	38.86	14.42	12.82	4.43	2.28	1.39	.82	136700
$\frac{11}{16}$×3⅝×6⅛	28	8.25	43.18	16.34	14.1	4.98	2.29	1.41	.84	150400
$\frac{3}{4}$×3½×6	29.3	8.63	42.12	15.44	14.04	4.94	2.21	1.34	.81	149800
$\frac{13}{16}$×3$\frac{9}{16}$×6$\frac{1}{16}$	31.9	9.4	46.13	17.27	15.22	5.47	2.22	1.36	.82	162300
$\frac{7}{8}$×3⅝×6⅛	34.6	10.17	50.22	19.18	16.4	6.02	2.22	1.37	.83	174900
$\frac{5}{16}$×3¼×5	11.6	3.4	13.36	6.18	5.34	2	1.98	1.35	.75	57000
$\frac{3}{8}$×3$\frac{5}{16}$×5$\frac{1}{16}$	13.9	4.1	16.18	7.65	6.39	2.45	1.99	1.37	.76	68200
$\frac{7}{16}$×3⅜×5⅛	16.4	4.81	19.07	9.2	7.44	2.92	1.99	1.38	.77	79400

Carnegie Steel Z Bars.—*Continued.*

Dimensions.	$w.$	$a.$	$I.$	$I'.$	$S.$	$S'.$	$r.$	$r'.$	$r''.$	$C.$
$\frac{1}{2}\times3\frac{1}{4}\times5$	17.9	5.25	19.19	9.05	7.68	3.02	1.91	1.31	.74	81900
$\frac{9}{16}\times3\frac{5}{16}\times5\frac{1}{16}$	20.2	5.94	21.83	10.51	8.62	3.47	1.91	1.33	.75	91900
$\frac{5}{8}\times3\frac{3}{8}\times5\frac{1}{8}$	22.6	6.64	24.53	12.06	9.57	3.94	1.92	1.35	.76	102100
$\frac{11}{16}\times3\frac{1}{4}\times5$	23.7	6.96	23.68	11.37	9.47	3.91	1.84	1.28	.73	101000
$\frac{3}{4}\times3\frac{5}{16}\times5\frac{1}{16}$	26	7.64	26.16	12.83	10.34	4.37	1.85	1.30	.75	110300
$\frac{13}{16}\times3\frac{3}{8}\times5\frac{1}{8}$	28.3	8.33	28.70	14.36	11.2	4.84	1.86	1.31	.76	119500
$\frac{1}{4}\times3\frac{1}{16}\times4$	8.2	2.41	6.28	4.23	3.14	1.44	1.62	1.33	.67	33500
$\frac{5}{16}\times3\frac{1}{8}\times4\frac{1}{16}$	10.3	3.03	7.94	5.46	3.91	1.84	1.62	1.34	.68	41700
$\frac{3}{8}\times3\frac{3}{16}\times4\frac{1}{8}$	12.4	3.66	9.63	6.77	4.67	2.26	1.62	1.36	.69	49800
$\frac{7}{16}\times3\frac{1}{16}\times4$	13.8	4.05	9.66	6.73	4.83	2.37	1.55	1.29	.66	51500
$\frac{1}{2}\times3\frac{1}{8}\times4\frac{1}{16}$	15.8	4.66	11.18	7.96	5.5	2.77	1.55	1.31	.67	58700
$\frac{9}{16}\times3\frac{3}{16}\times4\frac{1}{8}$	17.9	5.27	12.74	9.26	6.18	3.19	1.55	1.33	.69	65900
$\frac{5}{8}\times3\frac{1}{16}\times4$	18.9	5.55	12.11	8.73	6.05	3.18	1.48	1.25	.66	64500
$\frac{11}{16}\times3\frac{1}{8}\times4\frac{1}{16}$	20.9	6.14	13.52	9.95	6.65	3.58	1.48	1.27	.67	70900
$\frac{3}{4}\times3\frac{3}{16}\times4\frac{1}{8}$	23	6.75	14.97	11.24	7.26	4	1.49	1.29	.69	77400
$\frac{1}{4}\times2\frac{11}{16}\times3$	6.7	1.97	2.87	2.81	1.92	1.1	1.21	1.19	.55	20500
$\frac{5}{16}\times2\frac{3}{4}\times3\frac{1}{16}$	8.4	2.48	3.64	3.64	2.38	1.4	1.21	1.21	.56	25400
$\frac{3}{8}\times2\frac{11}{16}\times3$	9.7	2.86	3.85	3.92	2.57	1.57	1.16	1.17	.55	27400
$\frac{7}{16}\times2\frac{3}{4}\times3\frac{1}{16}$	11.4	3.36	4.57	4.75	2.98	1.88	1.17	1.19	.56	31800
$\frac{1}{2}\times2\frac{11}{16}\times3$	12.5	3.69	4.59	4.85	3.06	1.99	1.12	1.15	.55	32600
$\frac{9}{16}\times2\frac{3}{4}\times3\frac{1}{16}$	14.2	4.18	5.26	5.70	3.43	2.31	1.12	1.17	.56	36600

REINFORCED CONCRETE CONSTRUCTION.

A reinforced concrete construction is one where concrete and steel are used jointly, being proportioned to carry the strains of compression and tension respectively. Such constructions have all the advantages of a purely masonry construction along with the elasticity of one of steel. They are permanent, proof against fire, rust, rot, acid, and gas and do not require attention, repair, or painting. Moreover, the strength of concrete increases with age, and a safety factor of 4 at the time of completion of structure may easily amount to 6 or 7 after the lapse of a year or so.

Advantages. Crushed stone, sand, and cement are procurable on short notice, while structural steel is often subject to long delays in delivery. Concrete may be molded into any desired form, and masonry simulated. Deflection under safe load is practically nil. It being essential that a beam fail by the parting of the steel, after its elastic limit has been exceeded the stretch is such that a reinforced concrete beam should deflect several feet before failure.

Design. The concrete should be reinforced in both vertical and horizontal planes, the vertical reinforcement being inclined at an angle of 45° to the horizontal and approximating thereby the line of principal tensile stress. The shear members should be rigidly connected to the horizontal reinforcing steel. Steel should be distributed proportionally to the stress existing at any point.

The concrete should be composed of the best grade of Portland cement, sharp, clean sand and broken stone or gravel (to pass a 1-in. ring) in the proportions 1 : 2.5 : 5 for floor slabs and 1 : 2 : 4 for beams. Steel bars should be at least 0.75 in. from bottom of beam. The concrete should be thoroughly rammed into place and the centering left in position for at least 18 days, and, if freezing has occurred, for such additional time as may be required for every indication of frost to vanish and for the concrete to become thoroughly set.

Methods of Calculation. BEAMS. For span take dist. c. to c. of bearings (for slabs, the clear span + slab thickness; if continuous over more than one span, take dist. c. to c. of beams).

Maximum Bending Moments: Beam or slab supported at ends, $wl^2 \div 8$ (in.-lbs.); continuous or fixed beam, not less than $wl^2 \div 12$.

Coefficients of Elasticity: E_s (steel) = 30,000,000. E_c (concrete), for a

mixture not weaker than 1 : 2 : 4, may be taken as equal to 2,000,000. $E_s \div E_c = 15$.

That part of the concrete above the neutral axis is assumed to carry the compression, the stress varying uniformly from 0 at the neutral axis to a maximum at the compressed edge.

The steel reinforcement is assumed to carry all the tension, the stress being uniform over its cross-section.

Working Stresses:—If the concrete is of a quality such that its crushing strength after 28 days is 2,400–3,000 lbs. per sq. in., and the steel has a tenacity > or = 60,000 lbs. per sq. in., the following stresses may be allowed:

Concrete, in compression in beams subjected to bending	600 lbs. per sq. in.
Concrete in columns under simple compression	500 " " " "
Concrete in shear in beams	60 " " " "
Adhesion of concrete to steel	100 " " " "
Steel in tension	15,000 to 17,000 " " " "

For other concrete, allowable stress $= \frac{1}{4} \times$ crushing strength ($\frac{1}{5} \times$ crushing strength for columns). For steel of a different strength, allowable stress $= \frac{1}{2} \times$ stress at yield-point.

Let b = width of beam in inches.
d = depth of beam (top to center of steel bars) in inches.
b_1 = width of upper flange of T beam (slab) in inches.
d_1 = thickness of upper flange of T beam (slab) in inches.

(b_1 should not be greater than $\frac{1}{3}l$ or $\frac{3}{4}$ dist. c. to c. of floor-beams. d_1 is generally from $\frac{l}{12}$ to $\frac{l}{18}$.)

$A = bd$ = area of cross-section.
$m = E_s \div E_c$.
M = bending moment at section considered in inch-pounds.
t = tensile stress in steel in lbs. per square inch.
c = compressive stress in concrete in lbs. per square inch.
z = distance of resultant thrust in concrete from compressed edge of beam in inches.
kd = distance from compressed edge to neutral axis, in inches.
$A_c = kbd$ = area of concrete in compression in square inches.
A_t = area of steel in tension in square inches.
$p = A_t \div bd$ = ratio of steel to concrete = percentage of reinforcement.
l = span in inches.
w = load per inch run of span (weight of reinforced concrete may be taken as 150 lbs. per cu. ft.).

It can be shown that

$$k = \sqrt{p^2m^2 + 2pm} - pm, \tag{1}$$

and also that

$$bd^2 = \frac{M}{pt(1-\frac{1}{3}k)} \tag{2}$$

$$= \frac{2M}{kc(1-\frac{1}{3}k)}. \tag{3}$$

For T beams, where $d_1 > kd$,

$$A = b_1d_1 + b(d-d_1) = \frac{M}{pdt(1-\frac{1}{3}k)} \tag{4}$$

and

$$b_1d^2 = \frac{2M}{kc(1-\frac{1}{3}k)}. \tag{5}$$

When $d_1 < kd$,

$$A = \frac{M}{pt(d-z)} \tag{6}$$

and

$$b_1d_1 = \frac{2Mkd}{c(2kd-d_1)(d-z)}, \tag{7}$$

where

$$z = \frac{d_1}{3} \times \frac{3kd-2d_1}{2kd-d_1}. \tag{8}$$

Stirrups or rigidly attached web members or inclined bars should be provided at and near the ends of the beams where the average shearing stress on a vertical section exceeds 60 lbs. per sq. in. of the beam section.

The adhesion required in the case of a uniformly loaded beam, supported at the ends, with horizontal steel bars not turned up at the ends, is found as follows:

The difference of tension in 1-ft. length of bars at end of span = tangential force between the steel and concrete in that distance. At the end, $M=0$, and at 1 ft. from end $M=6w(l-12)$ in.-lbs. $\therefore$ increment of tension between the end and 1 ft. from the end is $6w(l-12)\div d(1-\frac{1}{3}k)$ lbs.

If n = total perimeter of reinforcing bars, the adhesion stress in lbs. per sq. in. $=w(l-12)\div 2nd(1-\frac{1}{3}k)$.

Example:—Design a beam, freely supported at the ends, of 12 ft. span to carry 960 lbs. per lineal foot, using 1% of reinforcement ($p=0.01$).

Taking $m=15$, $k=0.418$ [from (1)].

Assume $t=16{,}000$ lbs. per sq. in., $w=960\div 12=80$ lbs. per inch run, $l=12\times 12=144$ ins., $M=wl^2\div 8=207{,}360$ in.-lbs.

Substituting these values in (2),

$$bd^2=207{,}360\div[0.01\times 16{,}000(1-0.418)]=2{,}227.$$

For a trial value take $b=12$ ins. Then, $d^2=185$, and $d=13.5$ in. Adding $1\frac{1}{2}$ ins. to bottom of beam to protect the steel bars, gives a total depth of 15 ins.

A beam 12×15 ins. will weigh 15.6 lbs. per lineal inch, consequently w should be taken $=80+16$ say, and another calculation made, which will include the weight of the beam, approximately.

(The foregoing simple methods are those embodied in the Report of the Reinforced Concrete Committee, formed under the auspices of the Royal Institute of British Architects, and are undoubtedly as reliable as any that can be cited.)

Columns.—Let P = total weight to be supported by column in lbs.; and A_s = sectional area of steel reinforcement. Then, using the notation under "Beams":

For light longitudinal reinforcement (less than 1%), $P=c(A+mA_s)$. (1)
For heavy " " (1% and over), $P=c[A+(m-1)A_s]$. (2)

Example:—Design a column to sustain 40,000 lbs., using 1% reinforcement. $c=500$; $m=15$. Substituting in (2) ($A_s=0.01A$),

$$\begin{aligned}40{,}000&=500[A+(15-1)0.01A]\\&=500\times 1.14A=570A,\end{aligned}$$

and $A=70$ sq. ins. = 8.4 ins. square or 9.5 ins. diam.

The longitudinal bars should be bound around by steel straps or rods to prevent their bending, and these straps should not be spaced apart more than 24 times the least lateral dimension of the longitudinal bars.

Spiral Hooped Columns:—

$$P=0.6\left(1.178cd^2+\frac{7.54A_std}{s}\right),$$

where d = diam. of prism and hooping, A_s = sectional area of steel in the hoops or wire, and s = spacing of hoops or spirals—generally from $\frac{d}{10}$ to $\frac{d}{7}$. Take $c=400$ and $t=15{,}000$.

These formulas hold where the length of the column is not greater than 18 times the diameter or least lateral dimension. Few cases arise in practice where such a length is exceeded.

As the adhesion between the steel and concrete in the case of smooth bars is open to uncertainty, many bars have been devised which give a positive bond, through being twisted, corrugated, indented or otherwise deformed at close intervals along their length.

Summary of Beam Tests. From about 200 reported tests, T. L. Condron (W. Soc. of Engs., 3–15–03) deduces the following formula. Ult. B_m (in inch-lbs.) $=(nP+55)bd^2$, where $n=450$ for highly elastic steel bars positively bonded to the concrete ($=275$ for plain bars of ordinary structural steel); P = percentage of reinforcement = $(100\times\text{bar section})\div bd$; b and d in inches.

For ordinary concrete (1 3 6) P may vary from 0.5 to 1.25, economy lying between 0.7 and 0.9. For extra strong concrete (1 2 4) P may be increased to 1.25.

Stress Diagrams in Framed Structures. If three oblique forces maintain a body in a state of rest, their directions meet at one point and their proportional values may be shown by the respective sides of a triangle drawn parallel to the forces.

If a body remains at rest under the action of a number of forces in the same plane, their relative magnitude may be shown by a polygon whose sides, taken in order, are drawn parallel to the forces.

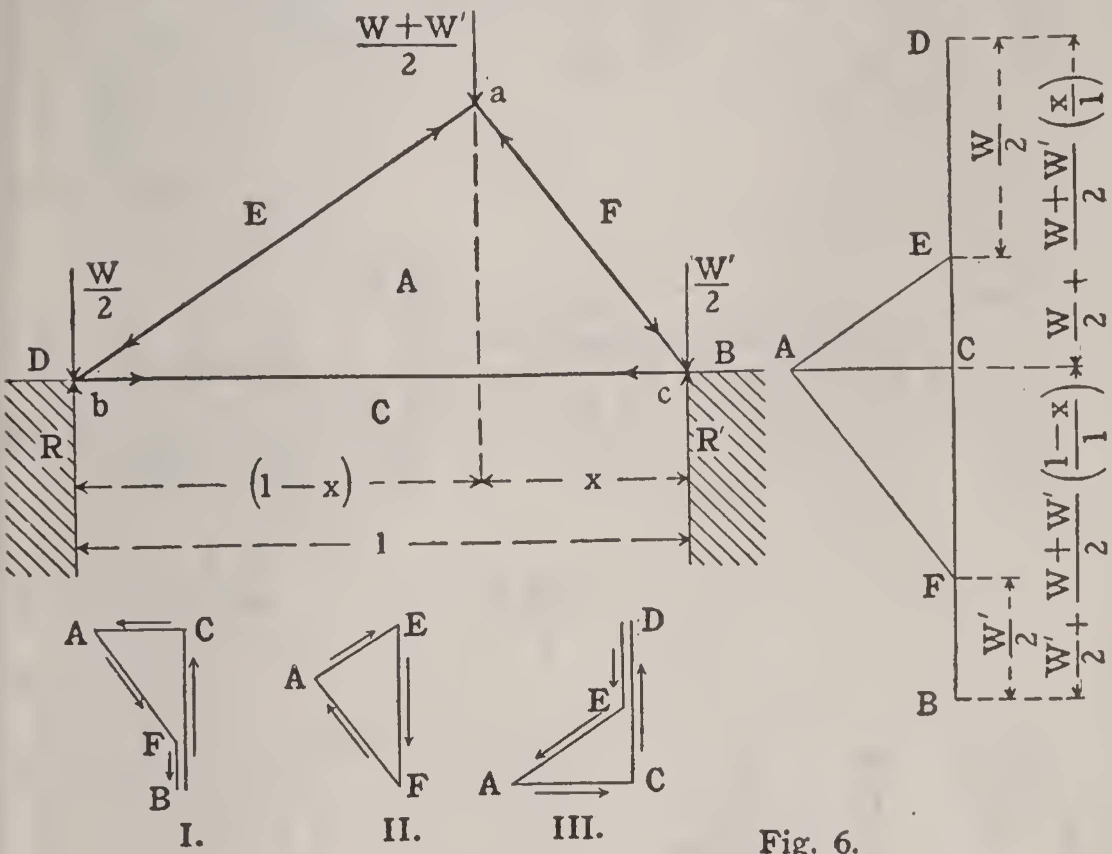


Fig. 6.

General Case. Simple Roof Truss (Fig. 6).

$\frac{1}{2}$ weight of $ab(W)$ will be supported at each point, a and b.
$\frac{1}{2}$ " " $ac(W')$ " " " " " " a " c.

The weight, then, at $a = \frac{W+W'}{2}$.

The reaction at R which balances $a = \frac{W+W'}{2} \cdot \frac{x}{l}$.

" " " R' " " $a = \frac{W+W'}{2} \cdot \frac{l-x}{l}$.

Total reaction at $R = \frac{W}{2} + \frac{W+W'}{2} \cdot \frac{x}{l}$.

" " " $R' = \frac{W'}{2} + \frac{W+W'}{2} \cdot \frac{l-x}{l}$.

The forces being thus stated, letter each cell or enclosed space (in this case but one, i.e., the triangle A), and also each section of the external space as divided by the lines of the forces and the members of the truss,

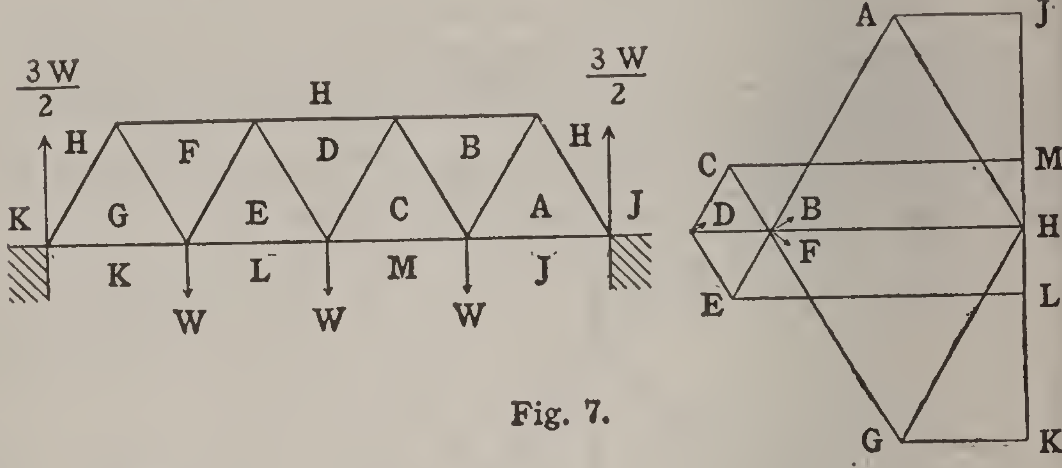


Fig. 7.

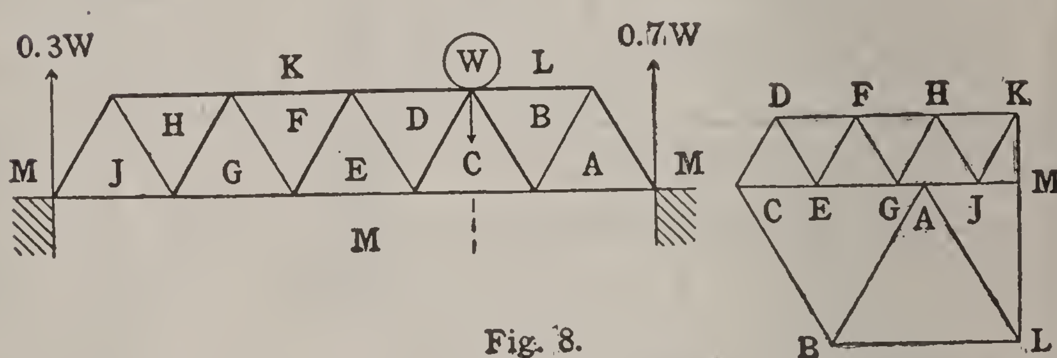


Fig. 8.

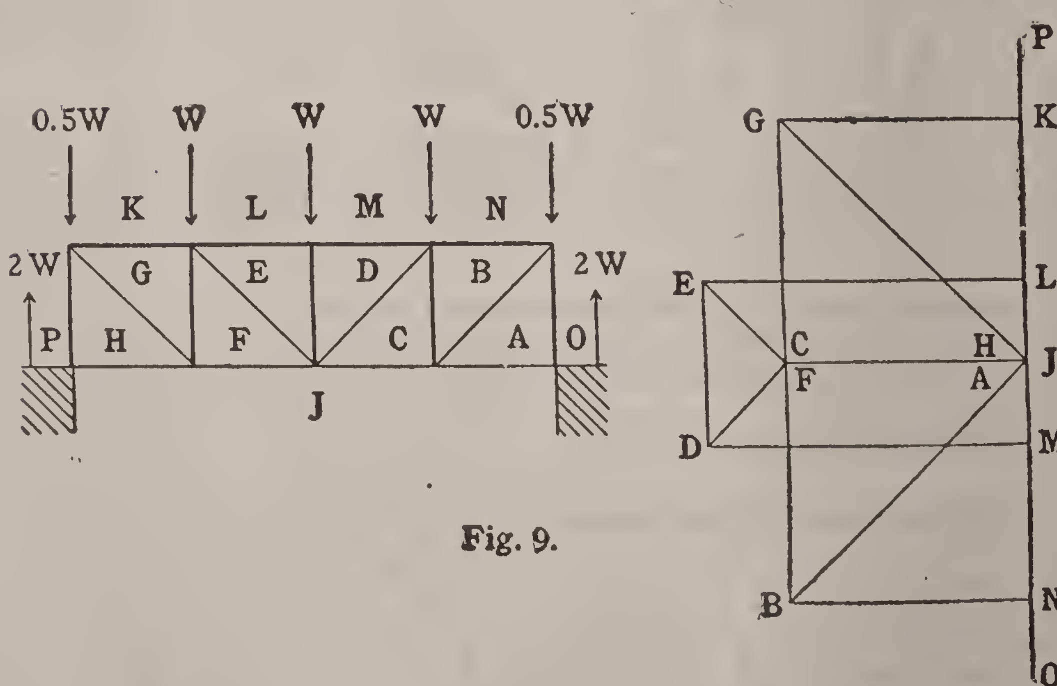


Fig. 9.

as B, C, D, E, and F. Draw the force diagram for each set of radiating forces. Consider the four forces at the point c, each defined by the spacial letters thus: FB, BC, CA. AF (using one direction of rotation through-

out, – preferably right-handed). Set off in the force diagram $FB = \frac{W'}{2}$ and $BC = R' = \frac{W'}{2} + \frac{W+W'}{2}\frac{l-x}{l}$. Draw AF parallel to the right member of truss, ac; then AC will be parallel to bc and meet BC at point C (see I). Notice that arrows must follow each other around the diagram in one direction. II and III show direction of forces for points a and b. AF,

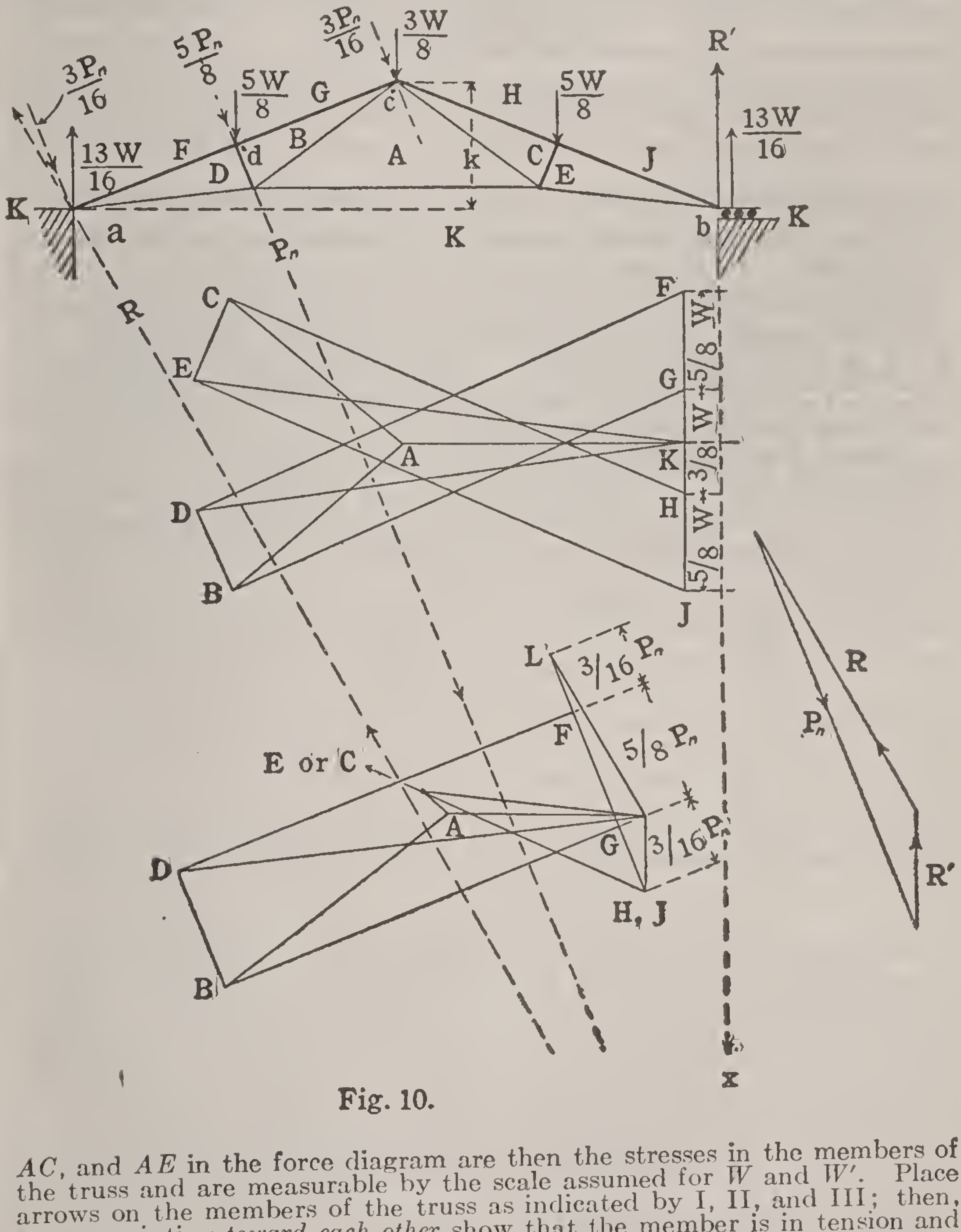


Fig. 10.

AC, and AE in the force diagram are then the stresses in the members of the truss and are measurable by the scale assumed for W and W'. Place arrows on the members of the truss as indicated by I, II, and III; then, arrows pointing *toward each other* show that the member is in tension and *vice versa* for compression. Generally $AE = AF$, $W = W'$, and $R = R' = W$.

The truss diagrams (Figs. 7, 8, 9, and 10) illustrate the application of the preceding principles. Redundant members (those not stressed excepting when distortion takes place) may be determined by inspection and their number = the number of members in excess of [(twice the number of joints) – 3].

Fig. 7 shows the stresses in a symmetrically loaded Warren truss, i.e., by the weight of its members. Fig. 8 shows the same truss under any concentrated load W, which may be taken for a rolling load by determining the stresses caused at each joint by imposing this load, and designing each member for the maximum stress it may have to withstand Note from BC, CD (Fig. 8), as compared with same members in Fig. 7, that the members are subject to either tensile or compressive stress and should be calculated for the greatest stress of each kind.

In the rafters of the roof-truss (Fig. 10) the load on each rafter $= W$, and, having three supports, is divided (as per table for Continuous Beams, *ante*) as follows: $\frac{3W}{16}$ at each end support and $\frac{10W}{16}$ on the middle support. The total horizontal wind pressure, P_t[$=40$ to 60 lbs. per sq. ft. $\times$ width of bay between two rafters $\times k$ (see diagram)] is resolved into two components,—one parallel, and one normal to the rafter. The latter, $Pn = \frac{P_t k}{ac}$ and is distributed at a, d, and c as $\frac{3Pn}{16}$, $\frac{5Pn}{8}$, and $\frac{3Pn}{16}$, respectively.

If a be fixed and b loose, expansion is provided for, and the reaction R' is vertical. R, R', and Pn mutually balance and meet in the point x (found by producing Pn to intersect R'). By connecting R and x the direction of R is given and values of R and R' are obtained from the auxiliary force diagram. If the wind blows from the right, Pn acts on bc, and x will be above instead of below b. Each member should be designed to resist the maximum stresses in it caused by the weight of roof, rafters, snow, and also the wind pressure, from whichever side a maximum stress in the particular member is caused.

Framed Structures of Three Dimensions must be solved by considering each plane of action separately. For example. in a shear legs substitute for the two rigidly attached legs a single one in a plane with the third or jointed leg, determine the respective stresses, and then resolve the stress in the substituted leg into the stresses for the two legs it replaces.

ENERGY AND THE TRANSMISSION OF POWER.

Force and Mass. The unit of force in engineering is one pound avoirdupois. Mass, or the quantity of matter contained in a body, $=\frac{\text{weight}}{g}$.

$$g=32.16954(1-0.00284\cos 2l)\left(1-\frac{2h}{r}\right),\text{ where}$$

$r=20{,}887{,}510(1+0.00164\cos 2l)$, in which l=latitude in degrees, h=height above sea-level in feet, and r=radius of the earth in feet. In calculations g is ordinarily taken as 32.16 in the U. S.

Velocity, or the rate of motion, is estimated in feet per second. If uniform, $s=\frac{v}{t}$. If uniformly varying from v_1 at beginning, to v_2 at the end of the time t, $s=\frac{v_1+v_2}{2}t$. (1).

Acceleration (f) is the increase of velocity during each second, and, if uniform, is produced by any constant force, the force being measured by the increase of momentum it produces. Momentum, or the quantity of motion in a body = mass × velocity = mv, and force producing acceleration $=wf\div g$. $f=\frac{v_2-v_1}{t}$ (2). Combining (1) and (2), $s=v_1t+\frac{ft^2}{2}$ (3). If $v_1=0$ (starting from a position of rest), $s=\frac{ft^2}{2}$ (4) and $f=\frac{v_2}{t}$ (5). Substituting (5) in (4), $v_2^2=2fs$ (6). For retarded motion (3) would read: $s=v_1t-\frac{ft^2}{2}$.

Impact of Inelastic Bodies. Two inelastic bodies after collision will move as one mass with a common velocity, and the momentum of their combined mass is equal to the sum of the momenta before impact. $(m_1+m_2)v(\text{final})=m_1v_1+m_2v_2$. $v=\frac{m_1v_1\pm m_2v_2}{m_1+m_2}$ accordingly as the bodies move in the same or in opposite directions before collision.

The Pendulum. A simple pendulum is a material point acted upon by the force of gravity and suspended from a fixed point by a line having no weight. A compound pendulum is a body of sensible magnitude suspended from a fixed point by a line or rod whose weight must be considered. The center of oscillation is a point at which, if all the weight of a compound pendulum be considered to be there concentrated, the oscillations will have the same periodicity as a simple pendulum. The distance of the center of oscillation from the point of suspension = (radius of gyration)^2 ÷ distance of center of gravity from point of suspension (a). An ordinary pendulum oscillates in equal times (isochronism) when the angle of oscillation does not exceed 5°.

Let l=distance in in. between point of suspension and center of oscillation of a simple pendulum, t=time in seconds for n oscillations, and n= number of single oscillations (one side to the other) in time t. Then, for a simple pendulum, $l=\frac{12gt^2}{\pi^2n^2}=\frac{39.1t^2}{n^2}$.

For a compound pendulum (rod of radius r): $l=\frac{4a}{3}+\frac{r^2}{4a}$;

" " " (ball of radius r): $l=a+\frac{2r^2}{5a}$.

" " " ball of weight W (dist. a) and ball of W_1 (dist. a_1), both on same side of point of suspension; $l=\frac{a^2W+a_1{}^2W_1}{aW+a_1W_1}$.

Balls $W(a)$ and $W_1(a_1)$, point of suspension between: dist. of c. of g. of system, $x=\frac{aW-a_1W_1}{W+W_1}$, and $l=\frac{a^2W+a_1{}^2W_1}{x(W+W_1)}$.

In the last two cases W is the larger weight, and the weight of connecting line or rod is neglected. The length of a simple pendulum which oscillates seconds at New York is 39.1017 in.

Energy, or the capacity for performing work, is of two forms: Potential Energy, which is stored or latent, and Kinetic Energy, or the energy of motion. In any system, kinetic energy + potential energy = a constant. In any machine the energy put in = the useful work given out + the work lost by resistances. (Stored energy not considered.) Either kind of energy may be transformed into the other kind.

Estimate of Energies. The Potential Energy of a weight w, at height $H=wH$ ft.-lbs. If allowed to fall, the velocity on reaching the ground, $v=\sqrt{2fs}$, from (6). But $f=g$, and $s=H$. $\therefore$ $v=\sqrt{2gH}$ and $H=\frac{v^2}{2g}$. Substituting (in wH), **Energy (now Kinetic)** in ft.-lbs. $=\frac{wv^2}{2g}$, which is applicable to *all* cases of moving bodies, it being strictly proper to assume that the velocity is caused by gravity.

When a body rotates around an axis (e.g., rim of fly-wheel, of weight, w), v (linear) $=2\pi Rn$, $\left(n=\frac{N}{60}\right)$ and the **Energy of Rotation** in ft.-lbs. $=\frac{wv^2}{2g}=\frac{w(2\pi RN)^2}{2g(60)^2}=0.0001704wR^2N^2$.

The Energy of a Compressed Spring $=\frac{wL}{2}$ ft.-lbs.; the **Energy of a Compressed Gas** = mean effective total pressure $\times$ stroke.

The Energy of One Heat Unit (1 B.T.U. = 1 lb. water raised 1° F. when near 39°) = 778 ft.-lbs.

Energy of Power Hammers. Energy of falling hammer $=\frac{wv^2}{2g}$. Energy received by the hot iron = mean total pressure in lbs. p, $\times$ depth of impression H, in feet, and $pH=\frac{wv^2}{2g}$. $\therefore$ $p=\frac{wv^2}{2gH}$. The greatest total pressure $=2p$.

Energy of Recoil. Let w_1 and w_2 = weights of gun (with carriage) and projectile; v_1 and v_2 = velocity of recoil and projectile velocity at muzzle. Then, $w_1v_1=w_2v_2$ and $v_1=\frac{w_2v_2}{w_1}$ The energy of a body in motion $=\frac{wv^2}{2g}$, hence the energy of recoil $=w_1\left(\frac{w_2v_2}{w_1}\right)^2\div 2g$, and the energy of the projectile $=w_2v_2{}^2\div 2g$.

Power is the rate at which work is performed, the unit being one horse-power, or 33,000 foot-pounds exerted during one minute.

Elements of Machines. A machine is an assemblage of parts whose relative motions are fully constrained, and its purpose is the transmission or the modification of power. Let P be the point where the power is applied and W the point where it is removed or utilized. Then, work put in at P = work taken out at W (neglecting resistances). As work = force $\times$ distance, $Ps=Ws_1$, or $\frac{W}{P}=\frac{s}{s_1}$, where s and s_1 are the distances traveled by P and W. Further,

$$\frac{\text{velocity of } P}{\text{velocity of } W}=\frac{\text{force } W}{\text{force } P}=\text{Mechanical Advantage, }\frac{W}{P}.$$

The Lever. By the principle of moments, $Pr = Wr_1$ and the Mechanical Advantage $= \frac{W}{P} = \frac{r}{r_1}$, r and r_1 being the respective radii of P and W from the fulcrum (for straight lever and parallel forces).

Lever Safety-Valve. Let w, w_1, and W be the weights of lever, valve, and ball, respectively in lbs., r, r_1, and R the distances from center of gravity of lever, valve center, and ball center to fulcrum, in in., d the valve diam., in in., and p the steam pressure per sq. in. of valve. Then,

$$W = \frac{(0.7854pd^2 - w_1)r_1 - wr}{R}.$$

If the lever is bent or the forces are not parallel, the arms r_1 and R are then equal to the length of the perpendicular drawn from fulcrum to the line of direction of each force.

Wheel and Axle. Mechanical Advantage $= \frac{r}{R} = \frac{\text{wheel radius}}{\text{axle radius}}$.

Train of Gearing. P is applied at radius of first wheel, transmitted by its toothed axle to circumference of second wheel which is toothed, by second axle circumference to third wheel circumference, etc.

Mechanical Advantage, $\frac{W}{P} = \frac{r_1}{R_1} \times \frac{r_2}{R_2} \times \frac{r_3}{R_3}$, etc.

Block and Tackle. The pull P on the rope through the distance s will raise the weight W through the distance

$$s_1 = \frac{s}{\text{No. of plies of rope shortened by the pull}}.$$

Mechanical Advantage $= \frac{W}{P} = \frac{\text{No. of plies shortened}}{1}$. In any movable pulley, $\frac{W}{P} = \frac{2}{1}$, W rising only one-half the height that P does.

Differential Pulley. Two pulleys whose diameters are d and d_1 rotate as one piece about a fixed axis. An endless chain passes around both pulleys and one of the depending loops of the chain passes around and supports a running block from which W is hung. P is applied on the chain running directly to pulley of larger diam., d.

$$\text{Mechanical Advantage} = \frac{W}{P} = \frac{P\text{'s dist.}}{W\text{'s dist.}} = \frac{\pi d}{\frac{\pi d - \pi d_1}{2}} = \frac{2d}{d - d_1}.$$

Inclined Plane and Wedge. While P moves through base b, W is raised through the height h, and Mech. Adv. $= \frac{W}{P} = \frac{b}{h}$. A cam is a revolving inclined plane.

The Screw is an inclined plane wrapped around a cylinder so that the height of the plane is parallel to the axis of cylinder. It is operated by a force applied at the end of a lever-arm (of length r) perpendicular to axis. Let p'' = pitch of screw = height of inclined plane for one revolution of screw. Then, Mech. Adv. $= \frac{W}{P} = \frac{P\text{'s dist.}}{W\text{'s dist.}} = \frac{2\pi r}{p''}$.

Connecting-Rods are subject to alternate tension and compression and the diam. d_1 at mid-length is calculated by means of Gordon's formula for columns (both ends hinged) where $r^2 = d_1^2 \div 16$, using a safety factor of 10 and values of a and b for steel. The diam. at small end (d) is designed to resist compression only, that at large end (d_2) being obtained by continuing the taper from small diam. to diam. at mid-length and thence to the large end, and is equal to $2d_1 - d$. Kent gives as the average of a large number of formulas considered by him: $d_1 = 0.021 \times$ diam. of cylinder $\times \sqrt{p\ (\text{steam})}$. Barr gives as the average of twelve Am. builders: $d_1 = 0.092\sqrt{\text{cyl. diam.} \times \text{stroke}}$ (for low-speed engines), and thickness, t (for rectangular sections, high-speed engines) $= 0.057\sqrt{\text{diam. cyl.} \times \text{stroke}}$ breadth $= 2.7t$. All dimensions in inches.

Connecting-Rod Ends. Strap-end: width $=0.8m$, thickness $=0.22m$ (increased to $0.33m$ at mid-length and also at ends when slotted for gibs and cotter); depth of butt-end of rod $=1.1d+\frac{3}{16}$in. $d=$diam. of crank-pin, $m=d+0.2$ in.

Crank-Arms (Wrought Iron). Hub diam. $=1.8d$; hub length $=0.9d$; diam. of crank-pin eye $=2d_1$; length of eye $=1.4d_1$; width of web $=0.75\times$ diam. of adjacent hub or eye; thickness of web $=0.6\times$length of adjacent hub or eye ($d=$least diam. of shaft; $d_1=$diam. of crank-pin).

Valve-Stems. Diam., $d_3=\sqrt{\dfrac{\text{total pressure on valve area}}{12,000}}$.

Eccentrics. Sheave diam. $=(2.4\times\text{throw})+(1.2\times\text{shaft diam.})$; breadth $=d_3+0.6$ in.; thickness of strap $=0.4d_3+0.6$ in. ($d_3=$diam. of valve-stem.)

SHAFTING.

For strength against permanent deformation, $d=3.33\sqrt[3]{\dfrac{\text{H.P.}}{N}}$. For stiffness to resist torsion (max. allowable twist $<0.075°$ per foot in length), $d=4.7\sqrt[4]{\dfrac{\text{H.P.}}{N}}$. These values are for W.I.; for steel shafts d has but 84% of the values given by formulas. In designing take the larger of the two values of d obtained from the formulas.

Average Practice. $d=\sqrt[3]{\dfrac{c.\text{H.P.}}{N}}$, where c (for cold-rolled shafting) for shafts carrying pulleys $=75$; for line shafting, hangers 8 ft. apart, $=55$; for transmission only, $=35$. For turned iron shafting under similar conditions multiply value of c by 1.75.

Length between bearings to limit deflection to 0.01 in. per foot of shafting: for bare shafts, L (in feet) $=\sqrt[3]{720d^2}$; for shafts carrying pulleys, $L=\sqrt[3]{140d^2}$.

Fly-wheel Shafts. For shafts carrying fly-wheels, armatures or other heavy rotating masses, find the equivalent twisting moment of the combined torsion and bending in inch-lbs. and apply same in the two formulas at the beginning of this topic, remembering that

$$\text{Twisting moment}=\frac{33,000.12}{2\pi}\cdot\frac{\text{H.P.}}{N}=63,025\ \frac{\text{H.P.}}{N}.\quad \text{(See p. 31.)}$$

Average Engine Practice. Crank-shaft diam., $d=6.8$ to $7.3\times\sqrt[3]{\dfrac{\text{H.P.}}{N}}$ for low and high speed respectively (Barr). Also, $d=0.42$ to $0.5\times$piston diam. (Stanwood). N for machine-shops $=120$ to 180; for wood-working shops, 250 to 300; for cotton and woolen mills, 300 to 400.

JOURNALS.

The allowable pressure p in lbs. per sq. in. on the projected area ($l\times d$) of journals is as follows: For very slow-speed journals, $p=3,000$; for cross-head journals, $p=1,200$ to 1,600; for crank-pin journals, low speed, $p=800$ to 900; ditto, Am. practice, 1,000 to 1,200; for marine engine crank-pin journals, 400 to 500; railway journals, 300; crank-pin journals for small engines, 150 to 200; main bearings of engine, 150; marine slide-blocks, 100; cross-head surfaces, 35 to 40 lbs. per sq. in.; propeller thrust-bearings, 50 to 70; main shafting in cast-iron boxes, 15.

Overhung Journals. On end of shaft. Constant pressure. When $N<150$, $d=0.03\sqrt{P}$ for W. I., and $0.027\sqrt{P}$ for steel; $\dfrac{l}{d}=1.5$ to 2. When $N>150$, $d=0.0244\sqrt{lP\div d}$ for W. I. and $0.019\sqrt{lP\div d}$ for steel. Also $\dfrac{l}{d}=0.13\sqrt{N}$ for W. I. and $0.17\sqrt{N}$ for steel.

Journals under Alternating Pressures (e.g., crank-pin). When $N<150$, $d=0.027\sqrt{P}$ for W. I. and $0.024\sqrt{P}$ for steel; $\frac{l}{d}=1$ for W. I. and 1.3 for steel. When $N>150$, $d=0.0273\sqrt{\frac{lP}{d}}$ for W. I. and $0.02\sqrt{\frac{lP}{d}}$ for steel; $\frac{l}{d}=0.08\sqrt{N}$ for W. I. and $0.1\sqrt{N}$ for steel. Am. Engine Practice: d (for crank-pin) $=0.22$ to $0.27\times$ piston diam.; $l=0.25$ to $0.3\times$ piston diam. (Stanwood). Cross-head pins: $d_1=0.8d$; $l_1=1.4d_1$.

Neck Journals, or those formed on the body of shaft need but two-thirds the diameter of overhung journals of the same length. For ball and socket shaft-hangers, $l=4d$; depth of shoulder on neck journal may be taken as $0.07d+\frac{1}{8}$ in.

Pivots. For $N<150$, $p=700$, 350, or 1,422 lbs. per sq. in., and $d=\sqrt{P}\times 0.05$, 0.07, or 0.035, for W.I., on bronze, C.I. on bronze, and W.I. or steel on lignum-vitæ, respectively. For $N>150$, $d=0.004\sqrt{PN}$ and $0.035\sqrt{P}$ for W.I. (or steel) on bronze and lignum-vitæ, respectively.

Collar Bearings. Outside diam. $D=\sqrt{d^2+\frac{\text{total thrust in lbs.}}{47\times\text{no. of collars}}}$. Thickness of collar $=0.4(D-d)=\frac{1}{2}\times$ space between collars. (d = shaft diam.).

Shaft Couplings. For a cast-iron keyed sleeve-coupling, $l=2.66d+2$ in.; external diam. of sleeve $=1.66d+0.5$ in. For a cast flange coupling, l of hub on each half $=1.33d+1$ in.; hub diam. $=1.66d+0.5$ in.; flange diam. $=2.5d+4$ in.; flange thickness $=0.166d+0.42$ in.; width of flange rim $=0.35d+0.86$ in.; no. of bolts $=2+0.8d$; diam. of bolts $=\frac{d}{8}+\frac{5}{16}$ in. For plates forged on abutting shaft ends, $t=0.3d$; outside diam. $=1.6d+(2.25\times$ bolt diam.); no. of bolts $=\frac{d}{2}$. (d = shaft diam.)

Brasses should have a thickness in the center (where wear is greatest) $=0.16d+0.25$ **in.**

BALL AND ROLLER BEARINGS.

Roller Bearings. Let n = number of rollers; d = diam. of rollers in in. (for conical rollers take diam. at mid-length); l = length of rollers in in.; then, if the rollers are sufficiently hard and are so disposed that the load is equally distributed over l and n, Load in lbs. $P=cnld$, where $c=355$ for C. I. rollers on flat C. I. plates, and 850 for steel rollers on flat steel plates (Ing. Taschenbuch).

Friction may be reduced 40 to 50% by the use of roller bearings.

The Hyatt flexible rollers consist of flat strips of springy steel wound spirally into tubular form; they give at all times a contact along their entire length. It is claimed for them that they save 75% of the lubrication (and 10 to 25% of the power) needed by ordinary bearings of equal capacity, and that they cannot become overheated.

Ball Bearings. Diam. of enclosing circle $=(d+c)F+d$, where d = diam. of ball; c = clearance between each pair of balls; F, a factor as follows:

No. of balls......	14	15	16	17	18	19
Factor F........	4.494	4.8097	5.1259	5.4423	5.7588	6.0756

No. of balls......	20	21	22	23	24
Factor F........	6.3925	6.7095	7.0266	7.3338	7.6613

or, generally, $D=d+\frac{d+c}{\sin\frac{180°}{n}}$, where n = no. of balls.

If $0.005n>\frac{d}{4}$, take $c=\frac{0.25d}{n}$; otherwise, $c=0.005$. All dimensions in inches.

Crushing Strength of Balls.

d	Breaking Load in Lbs. Ball on Ball.	Between Flat Plates.	Auto Machy. Co.	Safe Load.
$\frac{1}{8}$	1280	1814	1238	160
$\frac{1}{4}$	4153	6570	5150	640
$\frac{3}{8}$	9030	12700	11600	1450
$\frac{1}{2}$	16710	22610	20600	2570
$\frac{5}{8}$	28580	30000	32260	4030
1	59030	90650	82400	10300

The Auto Machinery Co.'s data answer to breaking load $=82{,}400d^2$ and are a fair average of the first two columns (results obtained by F. J. Harris at Rose Polytechnic institute), the surface of ball race being considered as between a spherical and a plane surface.

Greatest load on a single ball $= \dfrac{\text{total load} \times 5}{\text{No. of balls}}$ in an annular bearing where n ranges from 10 to 18 (Stribeck, Ing. Taschenbuch). Prof. C. H. Benjamin recommends a safety factor of 10, that in above table is 8.

Radial Ball Bearing, with 4 point contact. $P_{(\text{safe})}=(nd)^{3 \cdot 7}$. If $P>3{,}000$ lbs., $P=300+290nd$.

Thrust Bearing, with 3 point contact. $P_{(\text{safe})}(1{,}000 \text{ to } 4{,}500 \text{ lbs.})=1{,}143(nd-2\frac{1}{8})$; $P_{(\text{safe})}(4{,}500 \text{ to } 8{,}500 \text{ lbs.})=2{,}125(nd-4)$; $P_{(\text{safe})}(8{,}500 \text{ to } 17{,}000 \text{ lbs.})=1{,}500+808nd$.

Thrust Bearing, Balls between Flat Plates.

When nd	= 3	5	7	9	10
P, safe, in lbs.	= 475	1,200	2,200	3,200	5,000

Thrust Bearing, 2 Point (Balls in Races of Larger Diam.).

When nd	= 3	6	8	10	12	14
P, safe, in lbs.	= 300	800	1,500	2,750	4,000	4,800

Relation between Ball Diam. (d) **and Shaft Diam.** (D).

Three-point	Thrust Bearing,	$d=0.143+0.17D$
Flat-plate	" "	$d=0.125+0.19D$
Two-point race	" "	$d=0.0625+0.166D$
Radial, four-point.......	"	$d=0.3D$, when $D \leq 1.5$ in.
" "	"	$d=0.31+0.15D$ when $D>1.5$ in.

The foregoing proportion represents the practice of the American Ball Co., of Providence, as derived from their catalogue by the author and may be taken as guidance in design.

Friction of Ball Bearings. M. I. Golden (Trans. A. S. M. E.) from experiments on balls from $\frac{1}{4}$ to $\frac{1}{2}$ in. in diam. in radial or annular bearings at speeds from 200 to 2,000 r.p.m., deduces as a tentative formula

Friction = Load $\left(0.005+\dfrac{0.001}{d}+0.005D\right)$, where d = diam. of ball, and D = diam. of path of balls in the races.

At speeds around and exceeding 2,000 r.p.m. chattering takes place, which may be reduced to a marked degree by the use of oil. He found $\mu=0.00475$ (taken as 0.005 in formula).

Double Ball Bearings. In an ordinary ball bearing the turning of the shaft rotates the balls in such a manner that the surfaces of two contiguous balls rub or grind upon each other, and this is said to be the cause of a large proportion of the failures recorded in the use of ball bearings.

In the Chapman double ball bearing a smaller ball (not in contact with the shaft) is introduced between every two balls of the bearing proper, and a rolling contact throughout the bearing is thereby established. The Chapman Co. (Toronto, Ont.) claim to save 80% of the work lost in friction by ordinary self-oiling journal bearings, and refer to runs of $1\frac{1}{2}$ to 2 years duration without lubrication or appreciable wear.

GEARING.

Spur Gears are toothed wheels for transmitting power between parallel shafts, the teeth being parallel to the axes of the wheels. They are equivalent to friction cylinders or discs having teeth provided to avoid slipping with heavy loads and, with an infinite number of teeth, the gears would become smooth-surfaced cylinders engaging with each other at their circumferences. These circumferences are called pitch circles and the velocity relation between any two wheels is determined from their respective pitch circle radii.

For transmitting perfectly uniform motion the curves of the teeth are specially formed, the condition for such motion being that the normal to all surfaces of contact between the teeth must pass through the meeting-point of the two tangential pitch circles.

Epicycloidal Teeth for wheels are formed as follows: The part of tooth curve outside of the pitch circle is the path of a point on the circumference of an arbitrarily chosen circle which rolls on the outside of the pitch circle, and the part of tooth curve inside the pitch circle is the path of a point on the circumference of the same arbitrarily chosen circle when rolling inside the pitch circle.

For racks the pitch circle (of infinite diam.) becomes a straight line and the tooth outlines are generated by a point in the circumference of a circle rolling on the line above and below

Where gears are to work interchangeably the same rolling circle must be used throughout. Teeth should be designed so that at least two pairs are constantly engaged

Involute Teeth possess an advantage over epicycloidal teeth in that the distance between the wheel centers may be slightly varied without affecting the accuracy of contact; they are generated as follows: Draw pitch circles and connect their centers. Through point of contact of circles draw a line inclined at an angle of 75° to the line of centers and from each center draw a circle tangent to this line. These circles are base circles and the tooth curve in each wheel is the path made by a point in a line unwrapped from the base circle of that wheel. The prolongation of the outline inside the base circle to depth of tooth is a radial line. Diam. of base circle$=0.966\times$diam. of pitch circle. Involute rack teeth have straight outlines which make an angle of 75° with the pitch line.

Circular Pitch (p'') is the distance on the pitch line between the centers of two successive teeth. $p''=\dfrac{\pi\times\text{diam. of pitch circle}}{\text{No. of teeth}}$

Diametral Pitch (pd''), or the number of teeth per inch diameter of pitch circle$=\dfrac{\text{No. of teeth}}{\text{diam.}}=\dfrac{\pi}{\text{circular pitch}}$. Used largely in cut gearing.

Proportions of Teeth. If diam. of rolling circle for generating epicycloidal teeth is taken equal to $1.75\times$circular pitch, the tooth outline from pitch circle to bottom of tooth in a pinion of 11 teeth will be a radial line. Addendum (or radial height of tooth outside pitch circle)$=0.3p''$; Dedendum (or radial depth of tooth inside pitch circle)$=0.4p''$; hence, total length of tooth$=0.7p''$. Thickness of tooth on pitch circle$=\dfrac{19}{40}p''$; space between teeth$=\dfrac{21}{40}p''$; back-lash$=\dfrac{21-19}{40}p''=\dfrac{p''}{20}$; clearance $=(0.4-0.3)p''=\dfrac{p''}{10}$. The foregoing for cast wheels. For cut gears substitute 0.3, 0.35, 0.65, 0.485, 0.515, 0.03, and 0.05, respectively, for the above coefficients of p'' (Sellers).

Diametral Pitch Formulas for Small Gears (Brown & Sharpe Mfg. Co.). Let P=diametral pitch; D', d'=pitch circle diameters; D, d=outside diameters; N, n=nos. of teeth; V, v=velocity ratios (capitals for gear and small letters for pinion engaging with same); a=distance between centers of wheels; b=no. of teeth in both wheels. Then,

$$b=2aP;\ a=\frac{D'+d'}{2}=\frac{b}{2P};\ N=\frac{nv}{V}=\frac{bv}{v+V};\ n=\frac{NV}{v}=\frac{bV}{v+V}=\frac{PD'V}{v};\ V=\frac{nv}{N};$$

$$v=\frac{NV}{n}=\frac{PD'V}{n};\ D=\frac{2a(N+2)}{b};\ d=\frac{2a(n+2)}{b};\ D'=\frac{2av}{v+V};\ d'=\frac{2aV}{v+V}.$$

For a single wheel (in addition to foregoing lettering), let t=thickness of tooth or cutter on pitch circle; D''=working depth of tooth; s=addendum; f=amount added to tooth depth for clearance; $D''+f$=total depth of tooth; P'=circular pitch. Then, $P=\frac{N+2}{D}=\frac{N}{D'}=\frac{\pi}{P'}$; $P'=\frac{\pi}{P}$; $D'=\frac{DN}{N+2}=\frac{N}{P}$; $D=\frac{N+2}{P}=D'+\frac{2}{P}$; $D''=\frac{2}{P}=2s$; $N=PD'=PD-2$; $f=\frac{t}{10}$; $t=\frac{1.57}{P}=\frac{P'}{2}$; $s=\frac{1}{P}=\frac{P'}{\pi}=0.3183P'=\frac{D'}{N}=\frac{D}{N+2}$; $s+f=\frac{1}{P}\left(1+\frac{\pi}{20}\right)=0.3685P'$.

Bevel Gearing is used to connect shafts whose directions meet at any angle. The pitch surface of each gear is the frustum of a cone, both cones having a common vertex. The teeth have their surfaces generated by the motion of a straight line traversing the vertex while a point in the line is carried round the traces of the teeth on a conical surface, which surface is generated by a line drawn from the extremity of larger diameter of pitch surface frustum to the axis and perpendicular to an element in the pitch surface.

Spiral Gears are used to connect non-parallel shafts which do not intersect. Let α=angle of inclination of axes, and γ, v, n, R, N, t, and T be respectively the pitch angle, circumf. velocity, revolutions, radius, no. of teeth, circumferential pitch, and normal pitch of wheel A, and γ_1, v_1, n_1, etc., similar values for wheel B.

Then, $\gamma+\gamma_1+\alpha=180°$ and $\frac{v_1}{v}=\frac{\sin\gamma}{\sin\gamma_1}$, whence $\frac{n_1}{n}=\frac{R\sin\gamma}{R_1\sin\gamma_1}=\frac{N}{N_1}$. $T=t\sin\gamma$ and $T_1=t_1\sin\gamma_1$, and as T must equal T_1, $\frac{t}{t_1}=\frac{\sin\gamma_1}{\sin\gamma}$. For minimum sliding make $\gamma=\gamma_1$. The position of the common tangent at point of contact of the pitch cylinders is determined from $\frac{R}{R_1}=\frac{\cot\gamma}{\cot\gamma_1}=\left(\frac{n_1}{n}+\cos\alpha\right)\div\left(\frac{n}{n_1}+\cos\alpha\right)$. Also, $\cot\gamma=\frac{\sin\alpha}{\frac{n}{n_1}+\cos\alpha}$.

For $\alpha=90°$, $\frac{n_1}{n}=\cot\gamma$, or, $\frac{\text{revs. }(n)\text{ of follower}}{\text{revs. }(n_1)\text{ of driver}}=\tan\gamma$.

Worm Gearing. In this case $\alpha=90°$, $N=1$ and the teeth of B are inclined at an angle γ to the edge of wheel, and $\tan\gamma=\frac{t}{2\pi R}=0.15916\frac{t}{R}$.

Strength of Gear Teeth (Wilfred Lewis). Load in lbs. transmitted by teeth, $W=fp''by$, where b= width of tooth face, and y=a factor depending on the no. of teeth (n) and the curve employed.

y, for involute teeth, 20° obliquity $=0.154-\frac{0.912}{n}$;

" " " " 15° " (and epicycloidal) $=0.124-\frac{0.684}{n}$;

" " teeth with radial flanks $=0.075-\frac{0.276}{n}$.

Safe Working Stress, f, in lbs. per sq. in.

Speed of teeth in feet per min.		100	250	500	1,000	1,500	2,000	2,500
Steel,	f=	20,000	14,000	11,000	7,600	6,200	5,400	4,800
Bronze,	f=	15,000	10,500	8,200	5,700	4,600	4,000	3,600
Cast Iron,	f=	8,000	5,600	4,400	3,000	2,500	2,200	1,900

Approximate Strength. Safe load W, in lbs. $=300bp''$ for C. I. ($120bp''$, if shock is to be provided for. Lineham).

Rawhide. W in lbs. $=57$ to $114\times bp''$ (Ing. Taschenbuch). An American gear-maker, however, states that rawhide has the same strength as cast iron.

Bevel Wheels. $W=fp''by\times\frac{\text{small diam. of bevel}}{\text{large diam. of bevel}}$.

H.P. Transmitted $=(W \times$ velocity of teeth in feet per min.$) \div 33{,}000$.

Safe Maximum Speeds. 1,800 ft. per min. for teeth in rough, cast (iron) wheels; 2,500 ft. for cast-steel and 3,000 ft. for machine-cut cast-iron wheels.

Proportions of Gears. Face, $b=2p''$ to $2.5p''$; thickness of rim $=0.4p''+0.125$ in. at edge (add 25% for center); thickness of rim on bevel wheel (larger end) $=0.48p''+0.15$ in. (taper to vertex); width of oval arms (in plane of wheel) $=2p''$ to $2.5p''$; thickness of oval arms (parallel to shaft) $=p''$ to $1.25p''$, or half the width of arm; No. of arms $=0.55\sqrt{\text{No. of teeth} \times \sqrt[4]{p''}}$; taper of oval arms:—$2p''$ to $2.5p''$ wide at hub end tapered to from $1.33p''$ to $1.66p''$ at rim; thickness of hub $=p''+0.4$ in.; length of hub $=b$ to $1.25b$. For arms of cruciform section: width of webs in plane of wheel $=2p''$ to $2.5p''$; width of webs in plane of shaft $=b$ to $b+0.08p''$; thickness of webs in plane of wheel $=0.035p''$ (No. teeth $\div$ No. arms); thickness of webs in plane of shaft $=0.32p''+0.1$ in.

Driving Chain. Allowable velocities $=500$ to 600 ft. per min. No. of teeth in sprockets $=8$ to 80. Radius of sprocket $=p''\div 2 \sin(180° \div \text{No. of teeth})$. $p''=$ length of chord bet. centers of two adjacent teeth.

The Renold Silent Chain Gear consists of a chain made of stamped links of a peculiar form which runs on an accurately cut sprocket wheel. These links are joined by hardened-steel shouldered pins and are provided with removable split bushings. Advantages: high speeds (up to 2,000 ft. per min.); largest size (2 in. pitch, 10 in. wide) transmits 126 H.P. at 1,000 ft. per min.; positive velocity ratio; can be used on short centers, in damp or hot places, runs slack, thus obviating excessive journal friction; the contact is rolling instead of sliding and the running is practically noiseless. No. of teeth, 18 to 120. Where load or power is pulsating, a spring center sprocket is used to absorb the shock.

BELTING.

On account of slip, belting does not transmit power at an exact velocity ratio, but it is nearly noiseless and can be used over distances not exceeding 30 ft. without intermediate support.

Belt Tension. In any belt strained around a pulley and in motion there will be a slack side and a tight side. The tension on the tight side is equal to the tension on the slack side plus the frictional resistance to the slipping of the belt on the pulley. The relation between T_n (greater tension) and t_n (lesser tension) is: $\text{Log}\,(T_n \div t_n)=0.4343\mu l \div r=0.007578\mu\theta°$, where $l \div r=$ (arc of pulley embraced by belt) $\div$ (radius of pulley), and $\theta°=$ degs. of arc of pulley embraced by belt.

μ (coefficient of friction) for leather belts on iron pulleys $=0.3$ to 0.4 if dry, and 0.15 if oily; for wire rope, $\mu=0.15$ on iron pulleys and 0.25 on leather-bottomed pulleys; for hemp rope on iron pulleys, $\mu=0.18$ to 0.28.

The Driving Pull of a Belt $=T_n-t_n$, and the

Horse-Power Transmitted $=\left(\dfrac{T_n-t_n}{33{,}000}\right)V=\dfrac{(T_n-t_n)2\pi RN}{33{,}000}$.

Strength of Leather Belting. f_t (safe) $=320$ lbs. per sq. in. of section, which allows for lacing or other jointing (or, 275 lbs. for laced and 400 lbs. for lapped and riveted joints). Single belts run from $\frac{3}{16}$ in. to $\frac{5}{16}$ in. in thickness; double belts from $\frac{3}{8}$ in. to $\frac{3}{4}$ in. Section must be sufficient to meet T_n. Rubber belts: $f_t=11$ lbs. $\times$ No. of plies $\times$ width in in.

Tension in Belts due to Centrifugal Force (unimportant at low speeds). $f_t=\dfrac{12wv^2}{g}$ (where $w=$ weight of 1 cu. in. of leather $=0.0358$ lb.) $=0.0134v^2$, and total tension on tight side $=T_n+0.0134btv^2$.

Creep, Slip, and Speed. As the belt tension changes from T_n to t_n a slight retrograde movement, or creep, occurs which is due to the release of tension and which causes the follower pulley to revolve at a correspondingly decreased rate. This result is called the slip, and the loss amounts to about two per cent.

Belt Speed. Generally not in excess of 4,000 ft. per min., at which speed max. economy is shown. Belt speeds however rise as high as 6,000 ft. per min.

H. P. of Belting (approximate formula).

$$\text{H.P.} = \frac{\text{belt width in in.} \times \text{pulley diam. in in.} \times \text{revs. per min.}}{2{,}800}$$ for single belts. For double belts divide by 1,960 instead of 2,800.

Sag of Belts and Proper Distance between Shafts. (Sag in in. $= s$; Length in feet $= L$.)

Narrow belts over small pulleys, $L = 15$ ft., $s = 1.5$ to 2 in.; wider belts over larger pulleys, $L = 20$ to 25 ft., $s = 2.5$ to 4 in.; main belts over very large pulleys, $L = 25$ to 30 ft., $s = 4$ to 5 in.

Length of Belts. Open belt: $L = \pi(R + R_1) + 2\beta(R - R_1) + 2l \cos \beta$; Crossed belt: $L = 2(R + R_1)\left(\frac{\pi}{2} + \beta\right) + 2l \cos \beta$; where L = length of belt in in., R and R_1 = radii of larger and smaller pulleys, respectively, β = angle between straight part of belt and center line of pulleys (= No. of degrees $\times \pi \div 180$, in circular measure), l = distance between centers of pulleys in in.

Cone Pulleys (open belts). The length of belt must be the same for each pair of pulleys in the set, and the radii of the pulleys have the following relation: $RR_1 - (1.01414l + c)R_1 - (1.004724l + c)R = 0.51657l^2 - (1.01414l + c)(1.004724l + c)$. l being fixed by the design, insert values of R and R_1 for any one pair of pulleys and solve equation for c. Let the ratio of $R \div R_1$ for any other pair of pulleys $= n$. Substitute nR_1 for R, also value of c in equation and solve for R_1, taking the negative value of the root of right-hand member of the equation. This formula is absolutely accurate where $\beta < 30°$,—a limit including all practical applications. (For derivation see article by the compiler in Am. Mach., 5-19-04.)

Let n and n_1 be the lowest and highest respective speeds for any set of cone pulleys, and x the number of speed changes; then, the speed ratio between any two successive speeds, $a = \sqrt[x-1]{\frac{n_1}{n}}$, (geometric ratio). If a back-gear is used the number of speed changes is doubled and the speed ratio of the back-gear corresponds to the term of the series where it is introduced.

Principle in Belt Driving. The advancing side of belt must move at right angles toward the shaft it approaches, while the retreating side may make any deviation.

Lacing. Punch $b + 1$ holes in each end of belt, arranged zigzag in two rows. The edges of holes should be ¾ in. from sides and ⅞ in. from ends,—rows at least 1 in. apart. Lacing should not be crossed on the side running on pulley. (b = width in in.)

Cemented Belts. (Formula for canvas and leather.)

Gutta-percha, 16 parts; India rubber, 4; pitch, 2; shellac, 1; linseed-oil, 2; melt and thoroughly mix.

Leather-Belt Dressing. Use tallow for dry belts,—with the addition of a little resin for wet or damp places. For hard, dry belts apply neats-foot oil and a little resin. Oil drippings destroy the strength of leather. Leather should not be exposed to a temperature much above 110° F.

PULLEYS.

(Design of.) r = radius of pulley; b = width of rim = 1⅛ to 1¼ × width of belt; t = thickness of rim at center, = 0.2 to $0.25h$; t_1 = thickness of hub, = $0.75h$ to h; l = length of hub, = b; n = number of arms; h = width of arm at center of hub; h_1 = width of arm at rim, = $0.8h$; $n = 2.5 + \frac{r}{2b}$; $h = \frac{b}{4} + \frac{r}{10n} + 0.25$ in. Thickness of arms at hub and rim $= \frac{h}{2}$ and $\frac{h_1}{2}$ respectively. (The above for arms of oval cross-section.) Pulleys with more than one set of arms may be considered as separate pulleys combined, with dimensions for each as above, excepting that arm-proportions need be but from 70 to 80% of the values given. Crowning; rise at center of rim = $0.05b$.

Friction Gearing. P = total pressure forcing wheels together at line of contact; Fn = tractive force to overcome friction; μ = coefficient of friction, = 0.15 to 0.20, metal on metal; 0.25 to 0.30, wood on metal; 0.25, leather on iron; 0.2, wood on compressed paper.

$F_n = \mu P$; H.P. $= F_n V \div 33{,}000$. Transmits power without jar, but is limited to very light loads.

ROPE TRANSMISSION.

Wire Rope. Used where belting is impracticable, for spans of 70 to 400 feet. Ropes used are 6 strand, 7 to 19 wires per strand. The sheave pulleys have a deep V-groove with a rounded bottom of alternating leather and rubber blocks. The minimum diameters of sheaves for obtaining maximum working tension in rope without overstraining by bending are $150d$, $115d$, and $90d$, for ropes of 7, 12, and 19 wires per strand respectively, where d = diam. of rope in in. Actual H.P. transmitted $= 3.1d^2v$, where sheave diams. are $\geq$ above values. Proper deflection in feet = 0.0000695(span in feet)^2.

Speeds from 3,000 to 6,000 ft. per min. (v=ft. per sec.)

Manila Rope.

Diam. in in., d =	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$
Lbs. per 100 ft. =	9.5	16	20	30	34	42	50	70	112	130	170	192

Ultimate strength in lbs. $= 9{,}000d^2$. Safe tension, T_n on driving side $= 180d^2$ (lbs.). Centrifugal force, $F = \frac{wv^2}{g}$, where w = weight of 1 ft. rope. H.P. transmitted $= \frac{2vn(T_n - F)}{3 \times 550}$, where n = No. of wraps of rope around pulley. Best economical speed = 5,000 ft. per min. Add 250 ft. of rope to calculations to provide for tightener. Sheave dimensions: pitch diam. $= 40d$ to $80d$; outside diam. = pitch diam. $+ 2d + \frac{1}{16}$ in., center to center of grooves $= 1.5d$; center of groove to edge $= d + \frac{1}{16}$ in.

Cotton Driving Rope transmits about $\frac{1}{8}$ more power than Manila rope for the same diam. Sides of pulley groove are inclined at 45°; distance from center to center of grooves $= 1.5d$; width of groove at outside diam. $= 1.25d$. The bottom of groove is rounded with circle of diam. $= 0.66d$.

Sag, s (in inches) is obtained from the following formula: $T_n = \frac{wL^2}{8s} + ws$, for driving side, also $t_n \left(= \frac{T_n - F}{3} + F\right) = \frac{wL^2}{8s} + ws$, where L = length of span in feet.

FRICTION.

The tractive force necessary to overcome friction between the surfaces of solids depends (1) directly on the pressure between the surfaces in contact; (2) is independent of the area of the surfaces in contact, but increases in proportion to the number of pairs of surfaces; (3) is independent (at low speeds) of the relative velocity of the surfaces; (4) the tractive force depends on the coefficient of friction, μ, for the particular materials employed.

Tractive force, $F_n = \mu P$.

Coefficients of Friction, μ, for Plane Sliding Surfaces (Morin).

(For low speeds and light loads only.)

	Lubrication.						
	Dry.	Water.	Olive-oil.	Lard.	Tallow.	Dry Soap.	Polished and greasy.
Wood on wood	0.5	0.68	..	0.21	0.19	0.36	0.35
Metal on wood	.6	.65	.1	.12	.12	....	.1
Hemp on wood	.63	.87					
Leather on wood	.47	....	....	....	....	....	.28
Stone on wood	.6						
Stone on stone	.71						
Stone on W. I.	.45						
Metal on metal	.18	....	.12	.1	.11	....	.15
Leather on iron	.54						

Values of μ for Static Friction (Broomall).

	Dry.	Wet.		Dry.	Wet.
Steel on steel....	0.4408		C. I. on C. I......	0.3114	0.3401
Steel on C. I.....	.23		C. I. on tin.......	.454	
Steel on tin......	.365		C. I. on pine.....	.47	
Pine on pine.....	.474	0.635			

μ=tangent of the angle of friction, i.e., the greatest inclination possible before sliding occurs.

If surfaces are thoroughly lubricated the friction is neither solid nor fluid but partakes of the nature of both.

Comparison of Solid and Fluid Friction. Solid friction varies directly as the pressure and is independent of the area of surface and of velocity (when low). Fluid friction is independent of the pressure, varies directly as the area of wetted surface, directly as v (at very slow speeds), as v^2 (at moderate velocities) and as v^3 (at high velocities). For low speeds Morin's table may be used. For flat surfaces, 400 to 1,600 ft. per min., C. I. on C. I., lubricated, $\mu=0.23$, at a pressure of 50 lbs. per sq. in.

Friction of Journal Bearings (Beauchamp Tower). $\mu=c\sqrt{v}\div p$, where v=linear velocity in ft. per sec., and p=pressure in lbs. per sq. in. of the projected area of journal. (Projected area=length×diam.). Values of c vary according to the lubricant employed, viz.: Olive-oil, 0.289; lard-oil, 0.281; mineral grease, 0.431; sperm-oil, 0.194; rape-oil, 0.212; mineral oil, 0.276. These values are for thorough bath lubrication. To avoid seizing, p should not exceed 600 lbs. per sq. in.

A roughly accurate expression for the coefficient of friction for machinery oil, deduced from the experiments of Tower and Osborne Reynolds, is (for speeds over 20 ft. per min.):

$\mu=1.8\sqrt{v}\div \theta p$, where v= surface speed in ft. per min.,
θ= excess of oil temperature above 60° F., and p= bearing pressure in lbs. per sq. in.

The proper length for a lubricated bearing in which the bearing pressure shall be less than that required to squeeze out the oil film is $l=P\div 80(dN)^{\frac{1}{2}}$, where l is length in in., d= diam. in in., P= total load on bearing in lbs., and N=r.p.m.—(J. T. Nicolson and Dempster Smith, in *The Engineer*, London, Nov. 23, '06, and Feb. 8, '07.)

The following results were obtained by Prof. A. L. Williston (E. W. & E., 3-18-05).

	μ (average).	Pressure per Sq. In.
Hyatt Roller Bearing.....	.0118	80 to 345 lbs.
C. I. Bearing.............	.0608	80 to 250 "
Bronze Bearing...........	.112	80 to 145 "

The bearings were all $1\frac{1}{2}$ in. diam.×4 in., lubricated with moderately heavy machine-oil of good quality. The C. I. and bronze bearings were reamed to size and lapped to insure perfect surface and high polish. μ at starting for the Hyatt bearing was found to be 0.0058.

Friction of Collar Bearings. For $p=15$ to 90 lbs., $v=5$ to 15 ft., $\mu=0.036$.

Friction Loss in Journals and Collars (R=outside or mean radius for journal and collar, respectively). Work lost, in ft.-lbs. per min.= $FnV=\mu P\times 2\pi RN$, or, expressed in horse-power, H.P.$=0.0001904\mu PRN$.

Work Lost in Pivot Friction $=(0.5$ to $0.66)(2\pi RN\mu P)$ in ft.-lbs.

LUBRICATION.

Spongy metals like C. I., brasses, and white-metal alloys, lessen frictional resistance to a considerable degree, but the use of unguents is necessary for good results. Lubricants are solid, as graphite; semi-solid, as greases; liquid, as oils. The following are the best lubricants for the purposes indicated:

Low temperatures: light mineral lubricating oils.
Intense pressures: graphite or soapstone.
Heavy pressures at slow speeds: graphite, tallow.
Heavy pressures at high speeds: sperm, castor, or heavy mineral oils.

Light pressures at high speeds: sperm, olive, rape, or refined petroleum oils.

Ordinary machinery: lard-oil, tallow-oil, heavy mineral oil.

Steam cylinders: heavy mineral oils, lard, tallow.

Delicate mechanisms: clarified sperm, porpoise, olive and light mineral lubricating oils.

Metal on wood bearings. water.

Essential Properties of Good Lubricants. (1) Body or viscosity sufficient to prevent contact of surfaces. (2) Freedom from corrosive acids. (3) As much fluidity as is consistent with body. (4) Low coefficients of friction. (5) High flash and burning points. (6) Freedom from substances likely to cause gumming or oxidation

Specific Gravities of Lubricants. Petroleum, 0.866; sperm-oil, 0.881; olive- and lard-oils, 0.917; castor-oil, 0.966.

Flashing and Burning Points. Sperm-oil flashes at 400° F. and burns at 500° F.; lard-oil flashes at 475° F. and burns at 525° F.

Thorough lubrication (preferably the oil-bath) is essential in order to obtain the best results, and to prevent seizing.

Graphite as a Lubricant. Foliated or thin flake graphite when applied as a lubricant materially reduces friction and prevents seizing and injurious heating of bearings. It may be applied dry to surfaces where pressures are light, or mixed with oil or grease (3 to 8% graphite, by weight) for heavy pressures. It may also be used to advantage in the presence of high temperatures, as in steam, gas-engine, and air-compressor cylinders, and also in ammonia compressors and pumping-engines. Water of condensation often suffices for a mixing lubricant.

Graphite fills up the minute depressions and pores in metal surfaces, bringing them much nearer to a perfectly smooth condition so that a considerably thinner film of oil (which may have a greater fluidity than usual) will be sufficient.

A test of car-axle friction by Prof. Goss (bearing pressure 200 lbs. per sq. in.) gave the following results:

Sperm-oil only, 9 drops per min., rise in temp. per hour = 26° F.; $\mu = 0.284$
Sperm-oil with
4% of graphite, 12.9 " " " " " " " " = 28° F.; $\mu = 0.215$

(From catalogues of the Jos. Dixon Crucible Co.)

Power Measurement. Power is measured by dynamometers, which either absorb or transmit the power undiminished. The **Prony Brake** is the typical form of absorption dynamometer and consists of a horizontal lever connected to a revolving shaft or pulley in such a manner that the friction between the surfaces in contact tends to rotate the lever-arm in the direction of the shaft rotation. This tendency is resisted by weights on the lever-arm, and the weight that will just prevent rotation is ascertained. Let P = weight in lbs. on lever, L = length of lever in feet from center of shaft to point of application of weight, V = velocity in ft. per min. of point of application of weight if allowed to rotate at the speed of the shaft, N = r.p.m., and W = work of shaft or power absorbed per min.

Then, $W = PV = 2\pi LNP$ ft.-lbs., or, H.P. $= \dfrac{2\pi NLP}{33{,}000}$.

HEAT AND THE STEAM ENGINE.

Heat, according to the dynamical theory, is a mode of motion of the molecules of a substance, its intensity being proportional to the amount of motion and its most readily observed effect being that of the expansion of the substance.

Transfer of Heat. Heat will pass from the warmer of two bodies to the colder until their temperatures become equal, the transfer being effected by radiation, conduction, or convection.

Radiation is the transfer of heat from one body to another across an intervening medium whose temperature is not affected by the transfer. Dark, rough surfaces are the best radiators and are advantageous in apparatus for heating, while light, polished surfaces are the poorest.

Relative Radiating Values. Lampblack, 100; polished metals cast iron, 26; wrought iron, 23; steel, 18; brass, 7; copper, 5; silver, 3

Heat Units Radiated per Hour per Square Foot of Surface (for 1° F. difference in temperature). Polished metals: silver, 0.0266; copper. 0.0327; tin, 0.044; zinc and brass, 0.0491; tinned iron, 0.0859; sheet iron, 0.092. Other materials sheet lead. 0.133; ordinary sheet iron. 0.566; glass, 0.595; cast iron, new, 0.648; do., rusted, 0.687; wrought-iron pipe, 0.64; wood, stone, and brick, 0.736; sawdust, 0.72; water, 1.0853; oil, 1.48.

Conduction is the transfer of heat by contact between the molecules of a body or the surfaces of contact of two distinct bodies.

Relative Values of Good Conductors. Silver, 100; copper, 73.6; brass, 23.6; tin, 14.5; iron, 11.9; steel, 11.6; lead, 8.5; platinum, 8.4; bismuth, 1.8; water, 0.147.

Heat Units Transmitted per sq. ft. per hour, for 1° F. difference in temperature: copper, 643; brass, 557; W. I., 374; C. I., 316 (Isherwood). These values are for bright surfaces up to ⅜ in. thick. For surfaces coated with grease or saline deposits (i.e., condensers) Whitham states that these values should be multiplied by 0.323.

Relative Values of Poor Conductors as Heat Insulators: Mineral wool, 100; hair-felt, 85.4; cotton wool, 82; sheep's wool, and infusorial earth, 73.5; charcoal, 71.4; sawdust, 61.3; wood and air-space, 35.7.

Comparative Radiation from Covered Pipes. Bare pipe, 1.00; covering of magnesia+7% asbestos, 0.308; plaster of Paris+4% asbestos, 0.34.

Radiation from Bare W. I. Pipes in B.T.U. per sq. ft. per hour, per degree F. difference of temperature between pipe and surrounding medium (taken at 70° F.):

Degs. diff.	Radiation.	Radiation and Convection — Still Air.	Radiation and Convection — Moving Air.
10	0.743	1.247	1.583
50	0.816	1.55	2.038
100	0.911	1.773	2.344
150	1.035	1.983	2.615
200	1.167	2.18	2.856
250	1.22	2.4	3.12
300	1.32	2.6	3.37

Steam-Pipe Coverings. The following figures are for coverings 1 in. thick. (For each $\frac{1}{10}$ in. additional thickness (up to 1.5 in.) subtract the percentage given.) Under average conditions (air at about 70°, steam about

100 lbs. pressure) 1 sq. ft. of bare pipe will give off about 3 B.T.U. per hour.

B.T.U. radiated per hour per sq. ft. of surface for each deg. F. difference in temperature between steam and outside air (approx.):

Hair-felt, 0.375 (2.5%); Remanit, 0.415 (2%); Manville sectional, best, 0 5 (1.4%); Magnesia, 0.515 (5.2%); Asbestos sponge, felted, 0.575 (9.2%); Asbestos air-cell, 0.675 (12%); Navy asbestos, 0.7 (8.4%); Asbestos fire felt, 0.745 (11%).

Coverings of 85% magnesia and solid cork coverings (1 in. thick) save about 83% of the heat that would be radiated from a bare pipe. Remanit (carbonized silk, wrapped) saves about 87%.

Convection is the transfer and diffusion of heat in a fluid effected by the motion of its particles. Water in the bottom of a vessel, or air on the floor of a room, being heated, becomes lighter and rises, allowing colder fluid to take its place. Convection currents being thus formed the heat is distributed through the fluid.

Expansion results from the application of heat to all bodies. (For coefficients of linear expansion see foot of page 18.) Water between 32° and 39.1° F. is an exception to the general law: it contracts as the temperature increases. Cast iron, bismuth, and antimony expand when solidifying, while gold, silver, and copper contract.

Measurement of Heat. Temperature is a measure of the intensity of heat and is determined by the employment of a thermometer or a pyrometer.

Thermometers. The freezing- and boiling-points of water (under atmospheric pressure) are marked on all thermometers, the space between being graduated as follows:

System.	No. of Divisions.	Freezing-point.	Boiling-point.
Fahrenheit (F.).......	180	32°	212°
Centigrade (C.).......	100	0°	100°
Réaumur............	80	0°	80°

whence $F^\circ = 1.8\ C.^\circ + 32$ and $C.^\circ = \frac{5}{9}(F.^\circ - 32)$.

Pyrometers are used to measure very high temperatures, Le Chatelier's being a thermo-electric couple of platinum and platinum-rhodium alloy employed in connection with a galvanometer and calibrated scale. The high temperatures of furnaces may be approximately ascertained by means of the copper cylinder pyrometer. A small copper cylinder of weight w (specific heat = 0.0951) is allowed to attain the temperature t° of the furnace and then plunged into a known weight, w_1 of water whose initial and final temperatures are t°_1 and T° respectively. Then,

$$t^\circ = \frac{w_1(T^\circ - t_1^\circ) + 0.0951wT^\circ}{0.0951w}.$$

Heat Units. The heat motion in a body depends on its mass, heat capacity, and temperature.

The British Thermal Unit (B.T.U.) is the amount of heat required to raise the temperature of one pound of water through one degree Fahrenheit, the water being near the temperature of its greatest density, 39.1° F. One B.T.U = 778 ft.-lbs. of energy.

The Calorie (metric system) is the amount of heat required to raise one kilogram of water one degree Centigrade at or near 4° C. 1 B.T.U = 0.252 Calorie (Cal.). 1 Cal. = 3.968 B.T.U. 1 Cal. = 426.8 kilogram-meters = 3087.1 ft.-lbs.

Specific Heat. Bodies, weight for weight, vary in their capacities for absorbing heat. If the heat-absorbing capacity of water is taken as unity, the relative capacity of another substance is called its specific heat and is therefore equal to the amount of heat in B.T.U. required to raise the temperature of one pound of the substance through 1° F.

Specific Heats of Various Substances. Water at 39.1° F., 1.00; water at 212° F., 1.0132; ice at 32° F., 0.504; mercury, 0.0333; cast iron, 0.1298; wrought iron, 0.1138; steel, 0.117; copper, 0.0951; coal, 0.24; tin, 0.0562; lead, 0.0314; glass, 0.1976; brass, 0.0939; coal ashes, 0.215. Gases (under constant pressure) carbonic oxide, 0.2479; carbonic acid, 0.217; ammonia, 0.508; air, 0.2375; hydrogen, 3.409.

Expansion of Gases. Marriotte's Law. The volume of a given portion of a gas varies inversely as its pressure, if the temperature be con-

stant. $V \propto \frac{1}{P}$; $\therefore V = \frac{\text{a constant}}{P}$, and $PV = $ a constant. The pressure curve of a gas expanding according to this law is a rectangular hyperbola and is called the isothermal of the gas.

Gay-Lussac's Law. The increase in volume of a given portion of a gas varies directly as the increase in temperature if the pressure be constant. Let V, V_1, and V_2 be respectively the original volume, the increase in volume, and the final volume, and $t°$ the rise in temperature. Then, $V_1 \propto t°$, and $V_1 = V\alpha t°$, where α = coefficient of cubical expansion (= coeff. of linear expansion $\times 3$); $\therefore V_2 = V + V_1 = V + V\alpha t° = V(1+\alpha t°)$. α for air $= 0.00203611$ for each degree F.

Absolute Temperature. If a given volume of air at 32° F. be reduced 491.13° in temperature ($= 1 \div 0.00203611$), its volume will theoretically become zero and its heat-motion may be considered as having ceased. For a perfect gas, absolute zero is 492.66° F. below the melting-point of ice, or, practically, $-461°$ F. ($= -273°$ C.), from which point all temperatures should be reckoned. In reality, all gases liquefy before reaching absolute zero. Absolute Temperature (τ) = 461° + reading of thermometer in degs. F.

Combination of Marriotte's and Gay-Lussac's Laws. PV = a constant, and $PV \propto \tau$; $\therefore PV = R\tau$. For 1 lb. of air at 32° F. (12.387 cu. ft.) under atmospheric pressure (14.698 lbs. per sq. in. = 2,116.5 lbs. per sq. ft.), $PV = 12.387 \times 2{,}116.5 = 26{,}217.66$ ft.-lbs. $= R\tau$, and, as $\tau = 493°$, $R = 53.354$.

Latent Heat. In changing from solid to liquid and from liquid to gaseous states, bodies pass through critical points called respectively the points of fusion and of evaporation, and at these points heat is absorbed to perform the work of molecular rearrangement. The Latent Heat of a substance is the quantity of heat units absorbed or given out in changing one pound of the substance from one state to another without altering its temperature.

Latent Heat of Substances in B. T. U. per Lb. Fusion Ice, 142.6 to 144; iron, 41.4 to 59.4; lead, 10.55. Evaporation: Water, 965.7; ammonia, 529; bisulphide of carbon, 162 SO_2, 164.

Saturation and Boiling Points. Saturation is said to occur when all the latent heat required for steam has been taken up. Boiling occurs when the tension in the water overcomes the surrounding pressure. Dry saturated steam is that which has a specific volume, temperature and pressure corresponding to its complete formation. Wet saturated steam is that in process of formation and in contact with the water from which it is generated. Superheated steam is that which has its temperature raised above that of the formation point.

Specific volume = No. of cu. ft. per lb. Specific density = No. of lbs. per cu. ft.

Moisture in Steam is measured by a calorimeter, and the percentage of moisture, $w = 100 \times \frac{H - H_1 - k(T° - t°)}{L}$, where H = total heat, L = latent heat per lb. of steam at the pressure of the supply-pipe, H_1 = total heat per lb. at the pressure of the discharge side of calorimeter, k = specific heat of superheated steam, $T°$ = temperature of the throttled superheated steam in the calorimeter, and $t°$ = temperature due to the pressure on the discharge side ($t° = 212°$ F. at atmos. pressure and $k = 0.48$).

All but ½ to 1% of the moisture in steam may be removed by the use of a separator, in which apparatus the direction of steam flow meets with abrupt changes and the water particles by reason of their momentum are thrown out of the path of flow.

The Quality of Superheated Steam (or the percentage of heat in excess of that due to the pressure), $Q = [L + 0.48(T° - t°)] \div L$, where L = latent heat of 1 lb. of steam at the observed pressure, $T°$ = observed temperature, and $t°$ = temperature due to pressure.

Pressure and Temperature Relations of Saturated Vapor. Log $p = a + b\alpha^n + c\beta^n$ (Regnault).

	32° to 212° F.	212° to 428°		32° to 212°	212° to 428°
a	3.025908	3.743976	$\log \alpha$	9.998181–10	9.9985618–10
$\log b$	0.61174	0.412002	$\log \beta$	0.0038134	0.0042454
$\log c$	8.13204–10	7.74168–10	n	$t° - 32$	$t° - 212$

Rankine gives as a close approximation, $\log p = A - \frac{B}{\tau} - \frac{C}{\tau^2}$, where $A = 6.1007$, $\log B = 3.43642$, $\log C = 5.59873$, and $p =$ lbs. per sq. in. (in both formulas).

Sensible Heat,—Heat of the Liquid (h). The number of B.T.U. required to raise 1 lb. of water from the freezing-point to $t°$ Centigrade $= (t + 0.00002t^2 + 0.0000003t^3) \times 1.8$.

The Total Heat of Evaporation is the quantity of heat necessary to raise one pound of water from 32° F. to a given temperature and then evaporate it. Total heat (in B.T.U.) $= 1{,}091.7 + 0.305(t° - 32) = 1{,}081.94 + 0.305t°$. Latent heat = total heat − sensible heat = (approximately) $1{,}091.7 - 0.695(t° - 32)$. (For greater accuracy subtract the sensible heat as obtained from formula above from the total heat.)

Density (D), Volume (V), and Relative Volume (V_r) of Saturated Steam. The density or weight in lbs. of 1 cu. ft. of saturated steam may be obtained from $\log D = 0.941 \log p - 2.519$. The volume of 1 lb. of steam in cu. ft. may be obtained from $\log V = 2.519 - 0.941 \log p$. The relative volume or number of cubic feet of steam from 1 cu. ft. of water may be derived from $\log V_r = 4.31388 - 0.941 \log p$.

The External Work of 1 lb. of Steam, W_e (in B. T. U.) $= \frac{144p\text{ (cu. ft. in 1 lb. steam at } p, -0.016)}{778}$, where $0.016 =$ cu. ft. in 1 lb. of water.

Evaporation from and at 212°. In comparing the evaporative performances of boilers working under various pressures and temperatures, it is customary to reduce them to a normal standard efficiency expressed by the equivalent weight of water which would be converted into steam if it were supplied to the boiler at a feed temperature of 212° and evaporated at the same temperature and at atmospheric pressure. The equivalent weight of water evaporated "from and at" 212°, $W = \frac{H - h}{965.7}$, where $H =$ total heat of the steam generated at the given absolute pressure (gauge pressure + 14.7 lbs.) and $h =$ the heat of feed-water.

Properties of Saturated Steam. The following table is abstracted from the complete tables of Prof. C. H. Peabody, whose results are probably in more general use among engineers than any others. $H =$ total heat of the steam $= 1{,}091.7 + 0.305(t° - 32)$; $h =$ heat of the liquid; $L =$ latent heat of vaporization, $= H - h$. Internal work, $W_1 = L - \frac{pu}{778}$, where $u = v - .016 =$ increase of volume of water and steam during evaporation (1 lb. water = .016 cu. ft.). Entropy of liquid $\phi_w =$ specific heat $\times \log_e \frac{\tau}{\tau_0}$; entropy of vapor, $\phi_s = \frac{L}{\tau} + \phi_w$; $\tau = t° + 460.7$. p (absolute) = pressure above vacuum in lbs. per sq. in.; $v =$ vol. of 1 lb. of steam in cu. ft.; $w =$ weight of 1 cu. ft. of steam in lbs. The values above 325 lbs. pressure are from Buel's tables.

Cooling Water Required by Condensers. Heat lost by steam = heat gained by the water; or, lbs. steam × (sensible heat + latent heat − temp. of hot well) = lbs. water × (final temp. of water − initial temp. do.), which may be reduced to. lbs. water per lb. of steam, $w = (1\,113.94 + .305T_s - T_h) \div (T_w - t_w)$, where $T_s =$ temp. of steam at release, $T_h =$ temp. of hot-well (usually from 110 to 120° F.), T_w and $t_w =$ final and initial temps. of the cooling water.

This formula has been criticised by E. R. Briggs (Am. Mach., 5-18-05) because it assumes that the whole weight of entering steam must give up its heat of vaporization at the release temperature, when, as a matter of fact, some 20 to 30% of the steam is in the form of water at this point. He suggests the following formula which gives much smaller results. $w = \left(H - \frac{2{,}545}{x}\right) \div (T_w - t_w)$, where $H =$ total heat per lb. of steam supplied to engine (reckoned above T_h), $x =$ steam consumption of engine in lbs. per I.H.P. hour, and $2{,}545 =$ B.T.U. in one H.P. per hour.

Specific Heats of a Gas. The specific heat (k_p) at constant pressure of any normally permanent gas such as air is 0.2375 B.T.U.

Properties of Saturated Steam.

p (abs.).	$t°$ F.	$v.$	$w.$	$H.$	$h.$	$L.$
0.5	80	640.8	.00158	1106.3	48.04	1058.3
1	101.99	334.6	.00299	1113.1	70	1043.1
3	141.62	118.4	.00844	1125.1	109.8	1015.3
5	162.34	73.22	.01336	1131.5	130.7	1000.8
10	193.25	38.16	.02621	1140.9	161.9	979
14.7	212	26.42	.03794	1146.6	180.9	965.7
15	213.03	26.15	.03826	1146.9	181.8	965.1
20	227.95	19.91	.05023	1151.5	196.9	954.6
25	240.04	16.13	.06199	1155.1	209.1	946
30	250.27	13.59	.0736	1158.3	219.4	938.9
35	260.85	11.45	.08736	1161	228.4	932.6
40	267.13	10.37	.09644	1163.4	236.4	927
45	274.29	9.287	.1077	1165.6	243.6	922
50	280.85	8.414	.1188	1167.6	250.2	917.4
55	286.89	7.696	.1299	1169.4	256.3	913.1
60	292.51	7.096	.1409	1171.2	261.9	909.3
65	297.77	6.583	.1519	1172.7	267.2	905.5
70	302.71	6.144	.1628	1174.3	272.2	902.1
75	307.28	5.762	.1736	1175.7	276.9	898.8
80	311.8	5.425	.1843	1177	281.4	895.6
82	313.51	5.301	.1886	1177.6	283.2	894.4
84	315.19	5.182	.193	1178.1	285	893.1
86	316.84	5.069	.1973	1178.6	286.7	891.9
88	318.45	4.961	.2016	1179.1	288.4	890.7
90	320.04	4.858	.2058	1179.6	290	889.6
92	321.06	4.76	.2101	1180	291.6	888.4
94	323.14	4.665	.2144	1180.5	293.2	887.3
96	324.64	4.574	.2186	1181	294.8	886.2
98	326.12	4.486	.2229	1181.4	296.4	885
100	327.58	4.403	.2271	1181.9	297.9	884
102	329.02	4.322	.2314	1182.3	299.4	882.9
104	330.43	4.244	.2356	1182.7	300.9	881.8
106	331.83	4.169	.2399	1183.1	302.3	880.8
108	333.2	4.096	.2441	1183.6	303.8	879.8
110	334.56	4.026	.2484	1184	305.2	878.8
112	335.89	3.959	.2526	1184.4	306.6	877.8
114	337.2	3.894	.2568	1184.8	308	876.8
116	338.5	3.831	.261	1185.2	309.4	875.8
118	339.78	3.77	.2653	1185.6	310.7	874.9
120	341.05	3.711	.2695	1186	312	874
125	344.13	3.572	.28	1186.9	315	871.9
130	347.12	3.444	.2904	1187.8	318.4	869.4
135	350.03	3.323	.3009	1188.7	321.4	867.3
140	352.85	3.212	.3113	1189.5	324.4	865.1
145	355.59	3.107	.3218	1190.4	327.2	863.2
150	358.26	3.011	.3321	1191.2	330	861.2
155	360.86	2.919	.3426	1192	332.7	859.3
160	363.4	2.833	.3530	1192.8	335.4	857.4
165	365.88	2.751	.3635	1193.6	338	855.6
170	368.29	2.676	.3737	1194.3	340.5	853.8
175	370.65	2.603	.3841	1195	343	852
180	372.97	2.535	.3945	1195.7	345.4	850.3
190	377.44	2.408	.4153	1197.1	350.1	847
200	381.73	2.294	.4359	1198.4	354.6	843.8
210	385.87	2.19	.4565	1199.6	358.9	840.7
220	389.84	2.096	.4772	1200.8	363	837.8
230	393.69	2.009	.4979	1202	367.1	834.9
240	397.41	1.928	.5186	1203.2	371	832.2
250	400.99	1.854	.5393	1204.2	374.7	829.5
260	404.47	1.785	.5601	1205.3	378.7	826.6
275	409.5	1.691	.5913	1206.8	383.6	823.2
300	417.42	1.554	.644	1209.3	391.9	817.4
325	424.82	1.437	.696	1211.5	399.6	811.9
500	467.4	0.942	1.062	1224.5	443.5	781
1000	546.8	0.48	2,082	1248.7	528.3	720.4

The specific heat at constant volume (k_v) is less, no external work being performed, and is equal to 0.1689 B.T.U.

Expressed in foot-pounds, and using capitals for symbols,

$$K_p = 184.77 \text{ ft.-lbs., and } K_v = 131.42 \text{ ft.-lbs.}$$

The specific heat of a gas at constant pressure is the same at all temperatures. External work $= P(V_1 - V) = R(\tau_1 - \tau)$.

Total heat $= K_p(\tau_1 - \tau)$; $\therefore$ Internal work $= (K_p - R)(\tau_1 - \tau)$.

When a gas is heated at constant volume only internal work is done, consequently $K_p - K_v = R = 53.354$ ft.-lbs.

The Specific Heat of Superheated Steam at constant pressure is usually taken as 0.4805. Grindley states that it averages from 0.4317 (between 230° and 246° F.) to 0.6482 (between 295° and 311° F.). Assuming a straight-line equation between these values, Specific Heat of superheated steam, $k_p(\text{at } t°) = 0.3451 + 0.00333(t° - 212)$.

Griessman (Z.V.D.I., 12–26–03) gives $k_p = 0.375 + 0.002083(t° - 212)$. Prof. C. R. Jones (E. R., 7–16–04) gives $k_p = 0.462 + 0.001525p$, where $p =$ absolute pressure in lbs. per sq. in. H. Lorenz (Z.V.D.I., No. 32–04) employs the following formula, where k_p varies as the pressure and inversely as the cube of the absolute temperature: $k_p = 0.43 + 1{,}476{,}000 \frac{p}{\tau^3}$ (p in lbs. per sq. in.; $\tau = 461° + t°$ Fahrenheit).

By making fair suppositions as to the temperatures involved in Jones' experiments, his results agree fairly well with those of Lorenz. For low pressures the value of Regnault (0.4805) seems corroborated by these investigators, while for pressures around 120 lbs. a value of 0.6 may be taken.

K_p for superheated steam (when $k_p = 0.4805$) $= 373.83$ ft.-lbs., and $K_v = 288.05$ ft.-lbs. $K_p - K_v = 85.78$ ft.-lbs. and $K_p \div K_v = 1.3$. The total heat of superheated steam, $H_1 = H + k_p(t_s - t)$, where H is the heat at temperature t of the steam at saturation and t_s is the temperature attained in superheating.

Expansion Curves. Adiabatics and Isothermals. The area A included by the ordinates P and P_1, the axis of abscissas and the curve of formula $PV = P_1V_1 = C$ is: $A = PV \log_e (V_1 \div V) = R\tau \log_e (V_1 \div V) = R\tau \log_e r$, where $r =$ ratio of expansion. When the curve is of the form $PV^n = P_1V_1^n = C$, $A = (PV - P_1V_1) \div (n-1)$. $n = \gamma = (K_p \div K_v)$ of the substance employed in the expansion.

When a gas expands against a resistance it performs work which requires an expenditure of heat. If the gas itself yields this supply of heat its temperature is lowered and the expansion is called adiabatic and represented by $PV^n = C$. If the heat required during the expansion be supplied from an external source the temperature of the expanding gas remains constant and the expansion is termed isothermal ($PV = C$).

Various Expansion Curves. Isothermal of a perfect gas: $PV = C$. Adiabatic of a perfect gas: $PV^\gamma = C$. ($\gamma = 1.3$ for superheated steam and 1.408 for air—usually taken as 1.41). Expansion of dry, saturated steam without becoming either wet or superheated: $pV^{\frac{17}{16}} = 475$ (Rankine), or $(p + 0.35)(V - 0.41) = 389$ (Fairbairn). Adiabatic of saturated steam: $pV^n = C$, where $n = 1.035 + 0.1 \times$ dryness fraction, the dryness fraction being the weight of the steam after the water particles are subtracted, divided by the weight of both steam and water particles. $n = 1.135$ for initially dry steam (Zeuner) and $n = 1.111$ for steam containing 25% of moisture (Rankine).

(For additional relations between p, v, and τ see Compressed Air.)

Specific Volume of Dry Saturated Steam. $V = \frac{Lt°}{\tau p} + v$. Take $t°$ at 1°, find the *increase* of pressure p from tables for 1°. $v =$ vol. of 1 lb. of water, in cu. ft. $L =$ latent heat at $\tau°$ F. (absolute). in ft.-lbs.

Volume of Superheated Steam. If greater than that of saturated steam, $PV_{\text{sup.}} = 93.5\tau_{\text{sup.}} - 971P^{0.25}$ (Peabody).

Thermal Efficiency of Heat Engines. Efficiency $= \frac{\tau - \tau_1}{\tau}$, where τ is the absolute temperature at which the heat is received (which should be as near to that of the furnace or gas explosion as possible), and τ_1 the

absolute temperature of rejection of the heat, i.e., that of the condenser or the atmosphere. If τ_1 were absolute zero, the efficiency would be the maximum attainable. The difference, therefore, $(\tau-\tau_1)$, should be the greatest possible with available temperatures.

Causes of Energy Loss in Steam Engines. Steam is not supplied at the furnace temperature (the greatest cause of loss), and the temperature of rejection is higher than that of the cooling water in the condenser. Steam is not compressed from the condenser temperature to that of the furnace, only a small part being compressed to the temperature corresponding to boiler pressure. If the condensed steam is not returned to the boiler a corresponding weight of feed-water must be heated to boiler temperature. Initial condensation in the cylinder causes waste, only a portion of the steam so condensed being re-evaporated during the stroke, and the expansion is not adiabatic. Clearance in the cylinder requires an additional amount of steam for each stroke which performs no work during the full pressure period of the stroke. Water particles in the steam (due to boiler priming) pass into the condenser without performing work and also abstract heat from the cylinder in their attempt to vaporize. τ must not be high enough to burn the cylinder lubricant or the packing and τ_1 is limited by the temperature of available condensing water. Radiation, leakage of steam, receiver drop in compound engines, wire-drawing, and friction losses (both of steam flow and of the moving parts of the engine) are additional causes of loss.

Initial Condensation. When saturated steam is admitted to a cylinder which has been cooled to exhaust temperature, part of it condenses. After cut-off the condensation continues, but, as the cylinder and steam temperatures become more nearly equalized, the latent heat liberated during liquefaction causes a partial re-evaporation. The initial loss is considerable, and, being but partially recovered through the re-evaporation, a quantity of water is rejected at release, part of which evaporates during the exhaust and causes back-pressure.

Methods of Reducing Cylinder Condensation. If the engine has a high rotating speed the time of each stroke is too short to allow the temperature changes which cause condensation to take place. Clothing the cylinder with non-conducting materials is a partial means of prevention. The supply of heat from live steam to the walls of the cylinder by means of a surrounding jacket assists re-evaporation providing that the piston speed is low enough to permit the absorption of the heat. By compounding, the work is divided among 2 to 4 cylinders and the range of temperature in a single cylinder is comparatively small. The saving due to compounding results from the re-evaporation taking place earlier in the total expansion.

The use of superheated steam is the most effective preventive of condensation. Saturated steam is allowed to flow through a coil or other form of superheater, its temperature being there sufficiently raised by the heat of the furnace gases to keep it dry, or nearly so, during the stroke. Superheat cannot exceed 750° F., cylinder lubrication being impossible at higher temperatures; the best results, however, are obtained between 650° and 700°. With superheat the pressures do not need to be so high, 160 lbs. being ample excepting in the largest engines. A moderate superheat of 100° to 150° above boiler temperature aids, especially in long pipe transmissions, and effects a saving of 10 to 12%.

At 120 lbs. pressure, with 170° superheat, 18% of the steam-consumption has been saved in a triple-expansion engine. A saving of 50% has been recorded, but 15 to 25% more nearly represents average practice.

The following formulas approximately express the results of a large number of tests (S = saving in per cent):

$$S=5.17+0.083\times\text{degs. F. of superheat (for turbines);}$$

$$S=4+0.12\times\text{degs. F. of superheat (for reciprocating engines).}$$

Heating Surface of Superheaters. $A\text{ (in sq. ft.)}=\dfrac{0.48W(t_1°-t_2°)}{6(t_3-t_1)}$, where 0.48 = sp. heat of superheated steam, W = lbs. of steam to be superheated per hour (boiler temp., t_2), t_1 = temp. after superheating, t_3 = temp. of furnace gases (1,000° to 1,200° F.), 6 = B.T.U. transmitted per sq. ft. of heating surface per hour, where (t_3-t_1) = 400° to 500° F.

Leakage is nearly independent of speed of sliding surfaces, is proportional to difference of pressure between the two sides of valve, and is inversely as the overlap of valve. With well-fitting valves it may amount to over 20% of the entering steam, and rarely falls below 4%.

For an unjacketed cylinder with a given ratio of expansion, initial condensation (expressed as a percentage of the steam in the cylinder) diminishes with increase of initial temperature, while the total condensation per stroke increases with such temperature increase.

Re-evaporation for a given ratio of expansion is as great, and sometimes greater, without jackets as with them, showing clearly that the regenerative action of the cylinder walls with a given ratio of expansion is largely independent of their mean temperature. (Prof. Capper, in Report of Steam-Engine Research Com. of I.M.E., 1905.)

Calculation of Initial Condensation and Leakage.

$$\frac{\text{Steam not accounted for by indicator}}{\text{Indicated steam}}=\frac{c \log_e r}{d\sqrt{N}},$$

where r=ratio of expansion, c=6 to 8 for simple unjacketed engines, 4 for jacketed slide-valve engines, 2 to 4 for Corliss engines (jacketed and unjacketed, respectively), and 12 for very poor engines.

Indicator Diagrams. (Fig. 11.) The figure shows the indicator diagram of a simple condensing engine, ON being the vacuum line or line

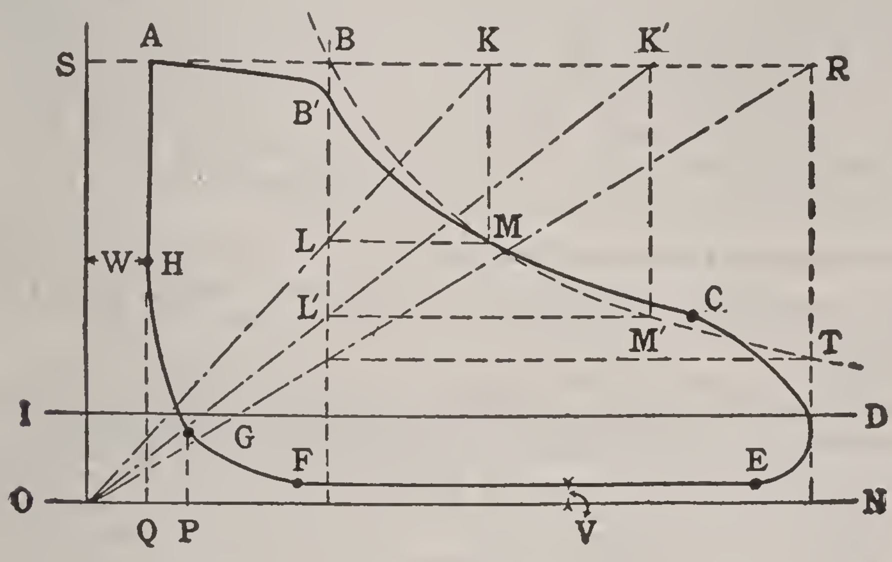


Fig. 11.

of zero pressure, OS the line of zero volume, and ID the atmospheric line of 14.7 lbs. absolute pressure (0 lbs. gauge). AR is the length of stroke and SA the clearance, which is the volume of the valve passages plus the volume between the piston at the end of stroke and the cylinder head reduced to a percentage of the stroke. (Clearance ranges from 2 to 7% of the total volume; when unknown it may be assumed as being 3% for well designed engines.)

The clearance space first fills, pressure rising immediately to A, and the piston moves to B, where the steam is cut off, and expansion takes place between B and C. If the cut-off is gradual (due to slow closing of the steam port), the steam will be "wire-drawn," and the pressure before cut-off will fall along the line AB'.

The exhaust port opens at C and the pressure drops to D and on the return stroke through D to E, where the port is fully open, and remains so until F is reached. The exhaust port closing at F, the remaining steam is compressed to G (cushioning the stroke), where incoming fresh steam, (due to the opening of steam-valve slightly before the commencement of the next stroke), rapidly raises the pressure to the starting-point A. The space V between the lines FE and ON represents the back-pressure

due to vapor pressure in the condenser, it being impossible to obtain a perfect vacuum. Back-pressure varies from 2 to 3 lbs. under fair conditions. The theoretical expansion curve BMT is an equilateral hyperbola (assuming the expansion to be isothermal) and should be drawn on the diagram or card for comparison. Taking any point M on the actual expansion curve $B'MC$, draw KM perpendicular to SR and intersecting it at K. Draw OK, and also ML parallel to SR and intersecting OK at L. Draw LB perpendicular to SR. B will be the theoretical point of cut-off. Any other point (M') may be determined by drawing OK'; then a perpendicular let fall from K' will intersect $L'M'$ (drawn parallel to SR from intersection of OK' and BL') at M', the desired point. Where the clearance is unknown it may be approximately fixed by selecting two points on the expansion line (B, M'), drawing the rectangle $BK'M'L'$ and producing the diagonal $K'L'$ to its intersection with ON at O.

Faults shown by Indicator Cards. (Fig. 12.) A,—too early admission; B,—too early release; C,—too early compression; D,—too late release; E,—too late admission; F,—too little compression; G,—too early cut-off; H,—choked admission; J,—choked exhaust; K,—leaky cut-off; L,—too much back-pressure; M,—double admission; N,—eccentric slipped backward; O,—eccentric too far ahead; P,—indicator inertia; Q,—sticking indicator piston; R,—initial condensation; S,—re-evaporation.

T shows the form of card obtained from gas-engines, the heavy line being the theoretical card. The explosive charge is drawn in along the atmospheric line, compressed along the lower curve, and ignited at the end of compression, when the pressure rises instantly. Expansion takes place along the upper curve to point of release, where the exhaust is then represented by the atmospheric line to the point of beginning of the cycle. In actual cards the ignition is not instantaneous but takes place along the dotted curves, the lower one indicating too late ignition and consequent loss of power. Release takes place before the end of the stroke, the pressure falling as shown by dotted line.

Calculation of Indicated Horse-Power. I.H.P. $=\frac{p_m La(2N)}{33{,}000}$, where p_m is the mean effective pressure throughout the stroke, in lbs. per sq. in., L=stroke, in feet, a=area of piston, in sq. in., and $2N$=No. of strokes per minute.

To obtain p_m (also abbreviated to *m.e.p.*), find the area of the card or diagram by means of a planimeter and divide same by its length, thus obtaining the mean (or average) height, and express this height in lbs. of pressure by comparison with the scale of the spring used in the indicator. Or, divide $ARNQ$ (Fig. 11) into 10 equal parts by vertical lines, measure the middle ordinate of each on the diagram, add same and divide by 10, thus obtaining the average height. Or, trace the card on section-ruled paper, ascertain the number of squares included by the boundary-line of the diagram and divide this number by the number of squares in one horizontal row between the extreme end ordinates of the diagram, thus obtaining the mean height. Should there be a loop in the diagram (as in Fig. 12 for too early cut-off) its area should be *subtracted* from the remainder of the diagram as the pressure indicated by the loop is negative.

Vacuum.—The best vacuum for a reciprocating engine is from 24 to 26 in. when the barometer is at 30 in.; with a better vacuum the additional gains are offset by the losses in obtaining same. A turbine should have the best obtainable vacuum, each additional inch above 24 in. reducing the steam consumption some 4 to 6%.

Indicated Water Consumption.—Lbs. water per hour per I.H.P. = $137.5[(b+c)w-cw_1]\div p_m$, where b=percentage of stroke completed at point where the calculation is made (which may be at any point between cut-off and release); c=percentage of clearance to the stroke; w=weight in lbs. of 1 cu. ft. of steam at the pressure of the point where the calculation is made; w_1=lbs. in 1 cu. ft. of steam at the final compression pressure.

Diagram Factor. In a theoretical diagram with admission at boiler pressure (p) up to the point of cut-off, expansion along a hyperbolic curve, release at the end of stroke, exhaust at back-pressure (p_b), and no compression, $p_m=\frac{p}{r}(1+\log_e r)-p_b$, where r=ratio of expansion = number

of volumes the original volume has expanded to, p and p_b being absolute pressures.

The actual p_m of an engine may be found by multiplying the right-hand member of the above equation by c, the diagram factor.

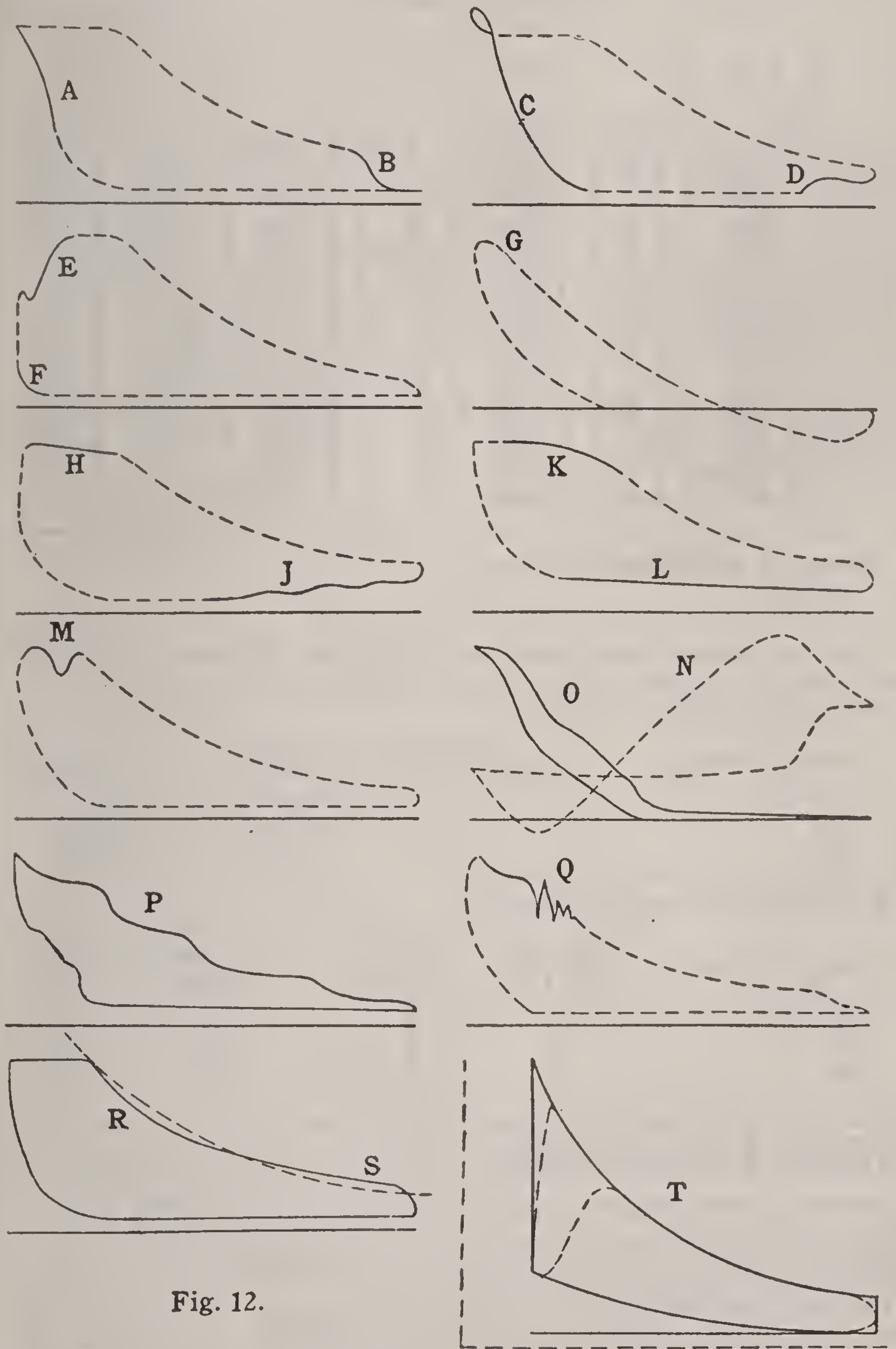


Fig. 12.

Values of c: 0.78 for simple, unjacketed, slide-valve engines. Compound engines,—0.6 to 0.8 for high-speed, unjacketed; 0.7 to 0.85 for

low-speed, unjacketed; slow-speed, jacketed, 0.85 to 0.9. Corliss, jacketed, 0.8 to 0.9. Triple-expansion,—high-speed, unjacketed, 0.7; marine engines, 0.6 to 0.66.

Hyperbolic Logarithms.

No.	Log.	No.	Log.	No.	Log.	No.	Log.
1	0	5.25	1.6582	9.5	2.2513	25	3.2189
1.25	.2231	5.5	1.7047	9.75	2.2773	26	3.2581
1.5	.4055	5.75	1.7492	10	2.3026	27	3.2958
1.75	.5596	6	1.7918	11	2.3979	28	3.3322
2	.6931	6.25	1.8326	12	2.4849	29	3.3673
2.25	.8109	6.5	1.8718	13	2.5649	30	3.4012
2.5	.9163	6.75	1.9095	14	2.6391	31	3.434
2.75	1.0116	7	1.9459	15	2.7081	32	3.4657
3	1.0986	7.25	1.9810	16	2.7726	33	3.4965
3.25	1.1787	7.5	2.0149	17	2.8332	34	3.5263
3.5	1.2528	7.75	2.0477	18	2.8904	35	3.5553
3.75	1.3218	8	2.0794	19	2.9444	36	3.5835
4	1.3863	8.25	2.1102	20	2.9957	37	3.6109
4.25	1.4469	8.5	2.1401	21	3.0445	38	3.6376
4.5	1.5041	8.75	2.1691	22	3.0911	39	3.6636
4.75	1.5581	9	2.1972	23	3.1355	40	3.6889
5	1.6094	9.25	2.2246	24	3.1781		

Diameter of Cylinder for any given I.H.P.

$$d = 144.9\sqrt{\text{I.H.P.} \div p_m LN}.$$

Cylinder Ratios for Multiple Expansion Engines.—For compound engines (2 cyls.), ratio $= \sqrt{\text{No. of expansions}} = 2.8$ to 3.5.

For triple expansion engines:

Gauge Pressure.	High Pressure Cyl.	Intermediate.	Low Pressure.
130 lbs.	1	2.25	5
140 "	1	2.4	5.85
150 "	1	2.55	6.9
160 "	1	2.7	7.25

For quadruple expansion engines:

Gauge Pressure.	High Pressure Cyl.	1st Intermediate.	2d Intermediate.	Low.
160 lbs.	1	2	4	8
180 "	1	2.1	4.2	9
200 "	1	2.15	4.6	10
220 "	1	2.2	4.8	11

The most economical point of cut-off in a simple, non-condensing engine lies between $\frac{1}{4}$ and $\frac{1}{5}$ of the stroke.

The Best Ratio of Expansion. The best number of expansions (N) in a simple condensing engine is $N = \frac{V_1 \tau_1}{V L_1}\left(\log_e \frac{\tau}{\tau_1} + \frac{L}{\tau}\right)$, where τ and τ_1 are absolute temperatures, V and V_1 are vols. in cu. ft. of 1 lb. of steam, L and L_1 are latent heats. V, τ, and L for the beginning and V_1, τ_1, and L_1 for the end of the expansion (Willans).

Combination of Multiple Expansion Diagrams. In order to compare the expansion with any desired theoretical curve, the several diagrams of the multiple expansion cylinders must be plotted on the same horizontal scale of volumes, clearances being added to the volumes proper. Any reference curve R may then be drawn. (Fig. 13).

Steam Consumption of Engines.

Type.	I.H.P.	Boiler Pressure, Lbs. per Sq. In.	Lbs. Steam per I.H.P. per Hour.
Non-Condensing:			
Common Slide-valve..........	25 to 100	80	33 to 40
Single-valve Automatic, high speed....................	50 " 150	80	32 " 40
Double-valve Automatic, high speed....................	50 " 150	80	30 " 35
Field, with superheat.........	136	113	(18.6)
Corliss, Automatic	100 to 200	75 to 90	22 to 27 (17.5)
Compound " , high speed..	100 " 250	110 " 120	25 " 27
Condensing:			
Corliss, Simple................	200 and up	80	18 to 20
Compound Automatic, high speed....................	200 to 500	110 to 120	17 " 19
Compound Schmidt (superheat)	75	180	(10.17)
" Corliss............	400 and up	110 to 135	13 to 17
" Leavitt............	640	135	(12.16)
" Bollinckx..........	300	90	(12.19)
Triple Expansion, Marine and Pumping..................	300 to 1,000	160 to 180	11.2 to 15
Triple Expansion, Sulzer.......	615	140	(11.85)
" " , Allis........	575	120	(11.68)
Quadruple Expansion.........	....	180 to 200	10 to 12
Rice & Sargent Cross-compound...	420	143.4	(9.56)
(Vacuum, 26.8 in., superheated to 443° F., Cyls., 16.07 in. and 28.03 in. ($r=3.04$))...........	....	(at throttle)	
			Lbs. per E.H.P. Hour.
Westinghouse - Parsons Turbine, (Vacuum, 28 in., superheat, 100° F., 3,500 r.p.m., full load)..	553	150	13.55
Same (superheat, 140° F., 1,500 r.p.m.)......................	2,030	150	12.66
Same (saturated steam, 1,500 r.p.m.)......................	2,030	150	14.7
(A gain of 14% by superheating. Consumption at half load is 9% greater.).....................			

The values in parentheses are some of the most economical results ever obtained. These figures may be expected from first-class designs: non-condensing, 25 lbs.; condensing simple, 18 lbs.; compound, 16 lbs.; triple expansion, 13.5 lbs.

The following are some recent economical results with saturated steam: Westinghouse-Parsons Steam Turbine (Dean & Main test), 600 H.P., saturated steam at 150 lbs., 28 in. vacuum: 125% load, 13.62 lbs. steam; 100% load 13.91 lbs.; 75% load, 14.48 lbs.; 41% load, 16.05 lbs.; average, 85% load, 14.51 lbs. steam per H.P.

850 H.P Rice & Sargent compound Corliss engine, 120 r.p.m.; cylinder ratio, 1 : 4; clearances 4% and 7%; live-steam jackets on cyl. head, live steam in reheater. For 600 H.P load (150 lbs., 28.6 in. vacuum, 33 expansions) Prof. Jacobus' test showed a steam consumption of 12.1 lb. per H.P hour. The cyl. condensation loss was 22% and the jacket consumption 10.7% of the total steam used.

250 H.P. Van den Kerchove compound engine, with poppet valves; 126 r.p.m., cylinder ratio, 1, 2.97: clearances 4%, jackets all over cylinder, no reheater. For 117 H.P. load Prof. Schröter's test showed a steam consumption of 11.98 lbs. per H.P. hour (150 lbs. pressure, 27.6 vacuum, 32 expansions). The cyl. condensation was 23.5% and the jacket consumption 14% of the total steam.

The most economical engine reported is a Cole, Marchent & Morley vertical cross-compound, with unjacketed cylinders and having a receiving reheater between. Nominal H.P.=500; cylinders, 21 and 36 in., stroke,

36 in. Boiler pressure, 114.5 lbs. gauge, temperature of steam, 726° F. (=378° superheat). R.p.m.=100.7; I.H.P.=145.5. Vacuum 26.5 in. Steam per I.H.P. per hour=8.585 lbs., and at 481 I.H.P., 9.098 lbs. The engine is supplied with drop piston valves, and has run successfully for

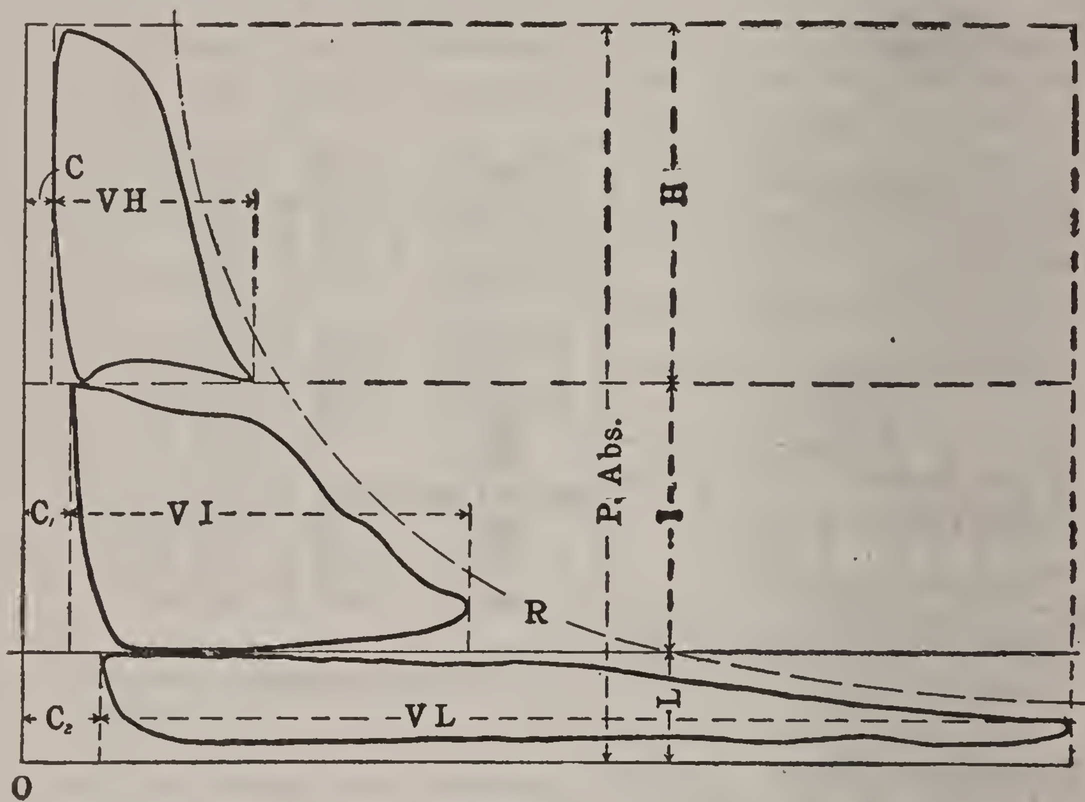


Fig. 13.

over a year, no trouble being experienced with the high temperatures employed. (*The Engineer*, London, June 2, 1905.)

Governors. Simple Fly-ball or Watt Governor. Let h=vertical distance from the point of support of the radius or pendulum arms to the plane in which the centers of gravity of the balls or weights revolve at any particular speed. Then, $h=\frac{35{,}200}{N^2}$ inches, and $N=\frac{187.6}{\sqrt{h}}$. Greater sensitiveness may be obtained by using the Porter type of governor, which has an axial weight w_1 in addition to the fly-ball weights (each=w) of a simple governor. In this case $h=\left(\frac{w+w_1}{w}\right)\frac{35{,}200}{N^2}$ in.

Valves. Zeuner's Diagram. When the crank is on the dead-center the normal slide-valve A should be at half-stroke, 90° in advance of the crank and on the point of admitting steam. If the valve has steam lap B added to it, the advance would necessarily be 90°+steam lap. To assist the steam under compression in cushioning the stroke, steam is admitted slightly before the end of stroke and at the dead-center the valve is then open by an amount called the lead, which must be added to the advance (90°+steam lap), to locate the position of the eccentric. Steam and exhaust laps (B and C) form an additional width to the valve-face and are for the purpose of effecting an early cut-off of steam or exhaust flow. (Fig. 14.)

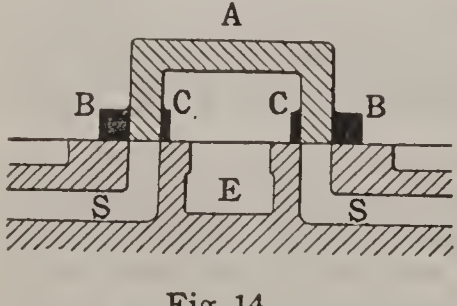


Fig. 14.

The action of a slide-valve is best shown by means of Zeuner's diagram (Fig. 15). On the diameter $AF(=2\times$eccentric throw) draw the circle $ABFH$. In the small diagram (I.) draw the steam-valve circle OF and also the exhaust-valve circle OA. With O as a center draw an arc with radius $OM(=$steam-lap) and also an arc with radius $OR(=$exhaust-lap) If the crank is on the dead-center A, the eccentric will be at B, or $90°+\theta$ in advance. The intercepts or shaded part MF made by the radius OB

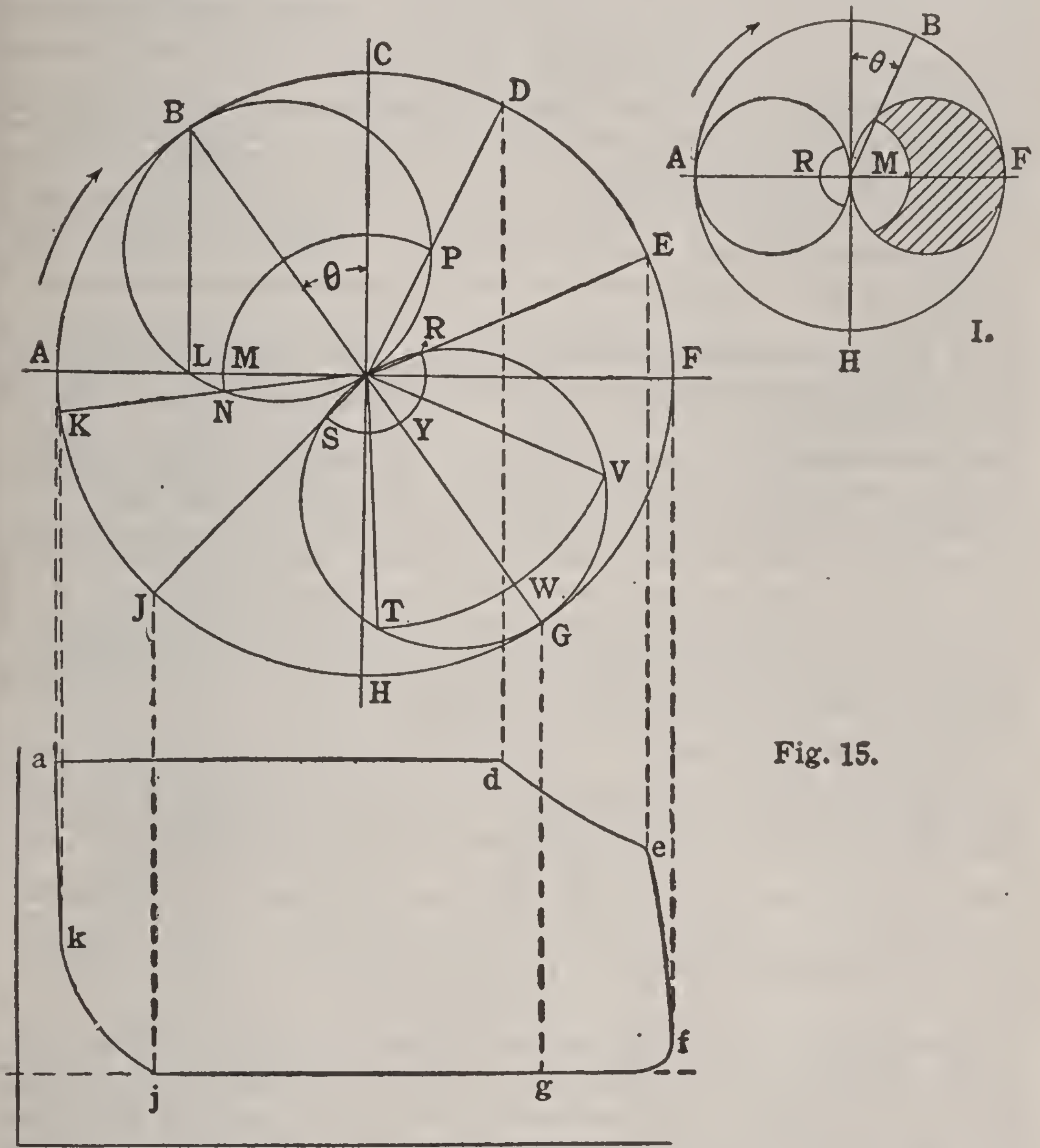


Fig. 15.

on the steam-valve circle will show the amounts of port opening for the corresponding positions of OB, or the eccentric.

The diagram may be used to better advantage by turning the valve-circles *back* $90°+\theta$, as in the main figure. Steam is admitted before the end of the previous stroke, the crank position being shown by OK which passes through the point N. The angle AOK is the angle of lead. At OA the crank is on a dead-center, at OB the steam-port is fully open and at OD steam is cut off by the closing of the port. From D to E the steam expands in the cylinder. At E the exhaust-port opens, reaching full opening at G and closing at J, the steam remaining in cylinder being compressed to K, where fresh steam is admitted for the next stroke.

OM is the steam-lap, OR the exhaust-lap, and LM is the linear lead due to the angular lead AOK. WY is the width of the steam-port and the exhaust has full opening from OV to OT. (O is center of circle ABF.)

By increasing the steam-lap, admission takes place later in the stroke and ceases earlier; expansion occurs earlier and ceases later; exhaust and compression are unchanged.

By increasing the exhaust-lap admission is unchanged, expansion begins as usual but continues longer, exhaust occurs later and ceases earlier, and compression begins earlier and ceases later.

By increasing the travel of the valve, admission begins earlier and ceases later, expansion occurs later and ceases earlier, exhaust begins and ceases later, and compression begins later and ends earlier.

By increasing the angular advance, admission, expansion, etc., all begin earlier but their respective periods are unaltered.

Valve Proportions. Ports should be dimensioned so as to allow a velocity of about 6,000 ft. per min. for live steam, and about 5,000 ft. per min. for exhaust. For a velocity of 6,000 ft. per min., Port area $= \frac{(\text{diam. of cyl.})^2 \times \text{piston speed}}{7{,}639}$. Length of port should be as near diam. of cyl. as possible, and width = area ÷ length. Width of exhaust port $= \frac{\text{travel}}{2}$ + width of steam-port − width of bridge between ports + exhaust lap.

For Corliss cylindrical semi-rotary valves; diam. of admission-valve = 3.2 × width of steam-port; diam. of exhaust-valve = 2.25 × width of exhaust-port. Length = diam. of cyl. Widths to be obtained from area formula for slide-valves.

Piston Speeds in Feet per Minute. Locomotives, 1,000 to 1,200; marine engines, 700; horizontal engines, 400 to 600; pumping-engines, 130. Cyl. area ÷ port area = 6,000 ÷ piston speed in ft. per min.

Flow of Steam. Lbs. per min. $= 0.85ap$ when discharging into the atmosphere. When flowing from one pressure to another which is d lbs. less and $p-d > .58p$, lbs. per min. $= 1.9ak\sqrt{(p-d)d}$. $k = 0.93$ for a short nozzle and 0.63 for an orifice in a thin plate (p = absolute pressure). Also, velocity in ft. per sec. $= 3.5953\sqrt{h}$, when h = height in feet of a column of steam of the given absolute initial pressure and of uniform density, whose weight is equal to the pressure on the unit of base.

Flow of Steam in Pipes. $v = 50\sqrt{\frac{HD}{L}}$, where L and D are the length and diameter of the pipe in feet and H is the height in feet of a column of steam at entrance pressure which would produce a pressure equal to the difference between the pressures at the ends of the pipe.

Q, in cu. ft. per min. $= 4.7233\sqrt{\frac{Hd^5}{L}}$, where d = diam. of pipe in inches

W, in lbs. flowing per min. $= 87\sqrt{\frac{w(p_1-p_2)d^5}{L\left(1+\frac{3.6}{d}\right)}}$ where w = lbs. per cu. ft. of steam at initial pressure, p_1, and p_2 = pressure at the end of pipe.

The Setting of Corliss Valves. There are three marks on the hub of the wrist-plate which indicate the extremes of throw and the central position accordingly as they coincide with another mark on the stand. Fix the wrist-plate in the central position, unhooking the rod connecting to the eccentric. Remove the back bonnets of the valves, and marks will be found on the valves and valve-chambers which indicate respectively the working edges of the valves and ports. By means of the adjustable rods which connect the valve-arms to the wrist-plate set the steam-valves so that they will have a lap of ¼ to ½ in. (the former for a 10-in. cyl., and the latter for a 35-in. cyl.,—intermediate sizes in proportion).

Similarly, set the exhaust-valves with $\frac{1}{16}$ to ⅛ in. lap for non-condensing, and with ⅛ to ¼ in. lap for condensing engines.

Adjust the dash-pot rods by turning the wrist-plate to the extremes of travel and regulate their lengths so that when they are down as far as they will go the steel blocks on the valve-arms will barely clear the shoulders on the hooks. (If the rods are too long they will be bent, if too short the hooks will not engage and the valves will not open.)

Hook the connecting-rod to the wrist-plate, loosen the eccentric, turn it over and adjust the eccentric-rods so that the wrist-plate will have correct

extremes of travel, as shown by the marks on hub. Place the engine on either dead-center, turn the eccentric enough more than one-fourth of a revolution in advance of the crank (in the direction of rotation) to show an opening of the steam-valve (at the piston end of cylinder) of $\frac{1}{32}$ to $\frac{1}{8}$ in., according to the speed, this being the lead. The higher the speed the more the lead required. Set the eccentric, turn to the other dead-center and obtain the same lead by adjusting the length of the rod connecting to wrist-plate. To adjust the regulator connections to the cut-off cams, turn the wrist-plate to one extreme of travel and adjust the rod connecting to the *opposite* cam so that the cam will clear the steel in the tail of hook by $\frac{1}{32}$ in. Turn to the other extreme of travel and adjust the other cam. To equalize the cut-off, block up the regulator about $1\frac{1}{4}$ in., which is its average position when running. Turn the engine slowly and note the positions of cross-head when the cut-off cams trip and the valves close. These positions should be at equal distances from the respective extremes of travel of the cross-head, and the rods should be adjusted until they are. Indicator cards should then be taken and such readjustments made as are required for the equalization of the diagrams.

To Place an Engine on a Dead-center. Locate by a mark on the guides the position of a mark on the cross-head when it is at any point near the end of the outward stroke. Denote this position on the fly-wheel rim by a mark which coincides with a fixed reference pointer. Turn the engine beyond the dead-center and on the return stroke until the mark on the cross-head coincides with that on the guides. Note this position on fly-wheel by making a mark at the reference pointer. Find the point midway between the two marks on the fly-wheel rim and turn the engine until this mid point coincides with reference pointer and the engine will be on a dead-center. To avoid the errors which might arise from looseness of bearings, the engine should be turned a little *beyond* the original position on the return stroke and the motion then reversed up to the original position so that the same brasses will press on the crank-pin in both observations.

Acceleration, Inertia, and Crank-effort Diagrams. The effect of the reciprocating parts of an engine is shown in Fig. 16. A vertical engine is chosen for illustration as both the inertia force and the dead weight of the moving mass are present, the effect of the latter being absent in a horizontal engine. Draw the crank-circle $JKLM$ with radius $0\,4 = 21$ in. and the connecting-rod $3\,4 = 90$ in. Draw the polar velocity curves KU and MU and also the velocity curve AXB. These curves are constructed as follows· In (II), if W moves uniformly, AW represents the crank velocity. Project the connecting-rod PW to C and AC will then be the corresponding piston velocity of the point P. Revolve AC to AE on the line AW and E will be a point in the polar velocity curve. Transfer AC to PF and F will be a point in the velocity curve JKH. The remaining points of each curve are similarly determined. The crank $0\,4$ makes 88 rev. per min., and the crank-pin consequently has a velocity of 16.1 ft. per sec. and OK (=ordinate X) should be divided into 16.1 parts to serve as a scale of measurement. The acceleration curve, QTR must then be drawn by the method shown in (III). Let AEB (III) be the velocity curve. Draw a tangent at any point E, a normal, ED and let fall a perpendicular EC to AB. Set off $CF = CD$ by revolving CD through 90° and F will be a point in the acceleration curve GKH. QT and TR show respectively the increase and decrease of velocity for the downward stroke and RT and TQ the acceleration and retardation for the up stroke.

The force moving the reciprocating parts around the dead-centers J and $L = \frac{wv^2}{gR}$. The inertia force, $= \frac{wf}{g}$, whence, f, the acceleration $= \frac{v^2}{R} = \frac{16.1 \times 16.1}{1.75} = 148$ ft. per sec. AQ, therefore, should be divided into 148 parts for a scale of acceleration in ft. per sec. The moving parts of the engine weigh 8,030 lbs. and the inertia force at any moment, $F = \frac{wf}{g} = \frac{8{,}030}{32.16} \times$ acceleration, or, at AQ (=148 ft. per sec.), $F = 36{,}911$ lbs. Draw NSP below QTR, each ordinate of distance between the two curves being equal to QN, which is 8,030 lbs. by scale where $AQ = 36{,}911$ lbs. NSP

is the curve of inertia pressure. The pressure per sq. in. on piston at $AQ = 36{,}911 \div 491 (= \text{area of piston in sq. in.}) = 75.2$ lbs. Draw the indicator cards to this scale, viz.: $EQXHB$ for the top of piston and $FPGA$ for

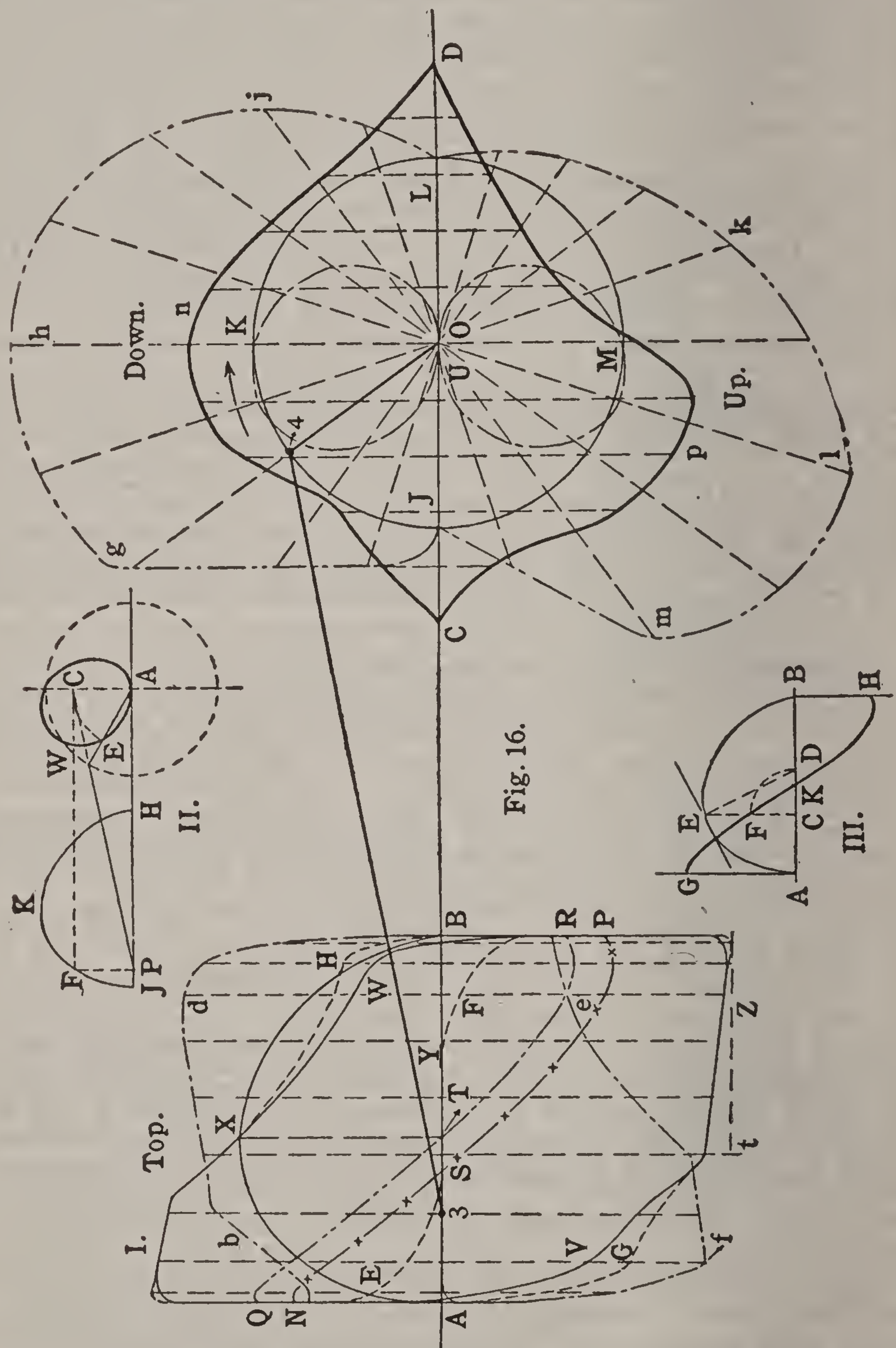


Fig. 16.

the bottom. When $QXHB$ is being drawn by the indicator on the top side of piston, AFR is being drawn on the bottom side, and, deducting the ordinates at F from those at H, the net effective pressure will be represented by the solid line WR. Similarly, by deducting E ordinates from

G the curve of net pressure is shown along VN The actual total pressure transmitted to the crank-pin during the first half of the stroke will be less than that shown on the indicator diagram by the amount required to set the reciprocating masses in motion, and during the latter half of the stroke the indicated pressure will be increased by the backward pull needed to absorb the inertia. The top card accordingly loses the area ANS and gains SBP, the resulting pressure areas then being $NIXWPSN$ for the top and $PZVNSP$ for the bottom, or, erecting the resulting ordinates on the base AB, the top and bottom areas are respectively $AbdBA$ and $BefAB$. To equalize these areas it will be seen that the cut-off on the bottom diagram is considerably later than that on the top diagram, on account of the dead weight which has to be supported. Only the reciprocating parts cause inertia force. The crank end of the connecting-rod is a rotating part, and it is customary to assume $\frac{2}{3}$ of the weight of the rod as reciprocating. The revolving parts are balanced by opposing weights on the crank-shaft. When the crank is on either dead-center all the pressure is received on the bearings, while at mid-stroke the pressure is exerted tangentially with no pressure on the bearings excepting that due to weight. At all other points the pressure is partly tangential and partly normal. The tangential pressure at any point is proportionally represented by the corresponding radius vector of the curve KU If JO is then divided into tenths the length of each radius vector in terms of these divisions will represent its virtual crank-arm in relation to the pressures transmitted along ABO. Multiply each net pressure ordinate along AB by its virtual crank-arm and set off the resulting tangential crank pressures radially, with the crank-circle $JKLM$ as a base line and the curves of crank-effort, $JghjL$ and $LklmJ$ will be obtained. These curves may be set out on a straight base by stepping JK out on CO, and KL on OD and then transferring the radial ordinates to vertical positions along the line CD when the curves CnD and DpC result. In locomotives two cranks at right angles are employed and in marine engines three cranks, 120° apart. A combination diagram may be made by superposing the diagrams of the individual cranks and adding the radial ordinates. (The foregoing discussion is taken from Lineham's Text-Book of Mech. Eng.)

Calculation of Fly-Wheels. On the base line EH (Fig. 17) lay out a series of crank-effort diagrams, making EAF and FCG equal to DpC and CnD of Fig. 16. $EG = 1$ rev. $= 3\frac{1}{2}\pi = 11$ ft. The mean ordinates of EAF and FCD are 29,500 lbs. and 25,000 lbs. respectively and one-half their sum, or 27,250 lbs., is the mean effort for the continuous diagram. Draw JK at this pressure above EH. The areas A, C, etc., above the

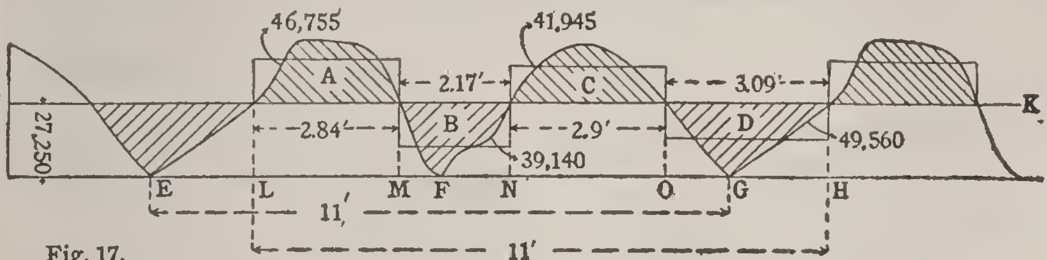


Fig. 17.

line JK show surplus work, while B, D, etc., below the line show deficits. The fly-wheel must absorb the work of A, C, etc., and give it out again at B, D, etc., thus tending to equalize the crank-effort. The mean pressures and distances are measured at A, B, C, and D and are shown by the work rectangles, and $A + C = B + D = 88{,}700$ ft.-lbs. The greatest rectangle is D, $= 49{,}560$ ft.-lbs., which is the amount of energy the fly-wheel must be able to deliver and thereby decrease its velocity. The heavier the wheel the smaller will be its fluctuation of velocity. Let $v =$ mean velocity in ft. per sec. and let the total fluctuation of velocity $= \frac{v}{k}$, where k varies from 20 to 300, according to the steadiness required. Let v_1 and v_2 be the maximum and minimum velocities at the mean radius,

E = the energy area (in this case 49,560 ft.-lbs.). Then $\frac{w(v_1^2 - v_2^2)}{2g} = E$, where w = weight of wheel in lbs. Now, $v_1 - v_2 = \frac{v}{k}$, $v_1 + v_2 = 2v$, and $v = 2\pi RN \div 60$, where R = radius of gyration of wheel in feet. Substituting and reducing, weight of wheel in lbs. $w = \frac{2{,}932Ek}{R^2N^2}$. Values of k $\left(\frac{1}{k} = \right.$ percentage of fluctuation from the mean speed$\left.\right)$.

For hammering and crushing machinery $k=5$; for pumping and shearing machinery, 20 to 30; for ordinary driving engines for machine-shops, 30 to 35; for milling machinery and gear transmission, 50; for spinning machinery, 50 to 100; for electric lighting, 150 to 300.

If the diameter of the wheel be large and the rim heavy (as compared with the arms and hub), R may be taken as the radius to center of rim section. If the hub and arms are of considerable weight, assume a section of fly-wheel, replacing the arms by a thin disc of equal weight and treat the whole cross-section of the wheel through the shaft as a beam section, finding its modulus, S, multiplying the same by y, the outer radius of wheel, and thus obtaining I, which divided by the total area of cross-section, will give R^2. v must be measured at R and great care taken to avoid the confusion incidental to calculating in both feet and inches.

$w = \frac{Cd^2s}{D^2N^2}$, where d, s, and D are diam. of cyl. in in., stroke in in., and diam. of fly-wheel in feet, respectively (J. B. Stanwood). Values of C: ordinary slide-valve engines, 350,000; Corliss engine for ordinary duty and slide-valve engines for electric lighting, 700,000; automatic high-speed and Corliss engines for electric lighting, 1,000,000.

Proportions of Steam-Engine Parts. In the following table the formulas attributed to Prof. John H. Barr are mean results obtained by him from some 160 engines (from 12 American builders) ranging from 20 to 750 H.P. Those of J. B. Stanwood are the conclusions of an extended practice and those of Wm. Kent are the best probable mean expressions of a large number of formulas considered and discussed by him in The Mechanical Engineer's Pocket Book. The following notation is employed: a = area of piston, l = length of stroke, d = diam. of piston, d_1 = diam. of fly-wheel, s = diam. of cylinder studs, t = thickness, l_1 = length of connecting-rod ($2.5l$ to $3l$). All in inch measure. N = r.p.m., p = max. steam pressure in lbs. per sq. in., V = piston velocity in ft. per min., H.P. and I.H.P. = rated and indicated horse-power, respectively. (See also related matter in Strength of Materials, *ante*.)

	Barr.	Kent.	Stanwood.
Cylinder:			
Thickness of walls,	$0.05d + 0.5$ in.	$0.0004dp + 0.3$ in.	
" " flanges,	$1.2 \times$ above		
" " heads,	" "	$0.00036dp + 0.31$ in.	
Studs, No. of (b),	$0.7d$	$0.0002d^2p \div s^2$	
" diam.,	$0.025d + 0.5$ in.	$0.01414\sqrt{p \div b}$	
Length of piston,	$0.46d$ (h.s.)	$\sqrt[4]{ld}$	
" " "	$0.32d$ (l.s.)		
Piston-rod diam.:			
High speed,	$0.145\sqrt{ld}$	$0.013\sqrt{pld}$	$0.14d$ to $0.17d$
Low "	$0.11\sqrt{ld}$		
Connecting-rods:			
High speed, rectangular section, thickness, t =	$0.057\sqrt{l_1d}$	$0.01d\sqrt{p} + 0.6$ in.	
Mean height =	$2.7t$	(Crank end, $2.25t$, cross-head end, $1.5t$)	
Low speed, circular section, mean diam. =	$0.092\sqrt{ld}$	$0.021d\sqrt{p}$	

	Barr.		Stanwood.
Cross-head pins: (L=length, D=diam.)			
High speed,	$LD=0.08a;\ \frac{L}{D}=1.25$		$L=0.25d$ to $0.3d$
Low "	$LD=0.07a;\ \frac{L}{D}=1.3$		$D=0.18d$ to $0.2d$
Crank-pins: (L=length, D=diam.)			
High speed,	$LD=0.24a;\ L=\frac{0.3\text{H.P.}}{l}+2.5$ in.		$L=0.25d$ to $0.3d$
Low "	$LD=0.09a;\ L=\frac{0.6\text{H.P.}}{l}+2$ in.		$D=0.22d$ to $0.27d$
Crank-shafts, Main Journals:			
High speed,	$\left(LD=0.46a;\ D=7.3\sqrt[3]{\frac{\text{H.P.}}{N}};\ L=2.2D\right)$		$L=0.85d$ to d
Low "	$\left(LD=0.56a;\ D=6.8\sqrt[3]{\frac{\text{H.P.}}{N}};\ L=1.9D\right)$		$D=0.42d$ to $0.5d$
Steam-ports, area:			
Slide-valve,	....		$0.08a$ to $0.09a$
High speed,	$aV\div5{,}500$		$0.1a$ to $0.12a$
Corliss,	$aV\div6{,}800$		$0.07a$ to $0.08a$
Exhaust-ports, area:			
Slide-valve,	....		$0.15a$ to $0.2a$
High speed,	$aV\div5{,}500$		$0.18a$ to $0.22a$
Corliss,	$aV\div5{,}500$		$0.10a$ to $0.12a$
Steam pipes, area:			
Slide-valve,	....	diam.=	$0.25d+0.5$ in.
High speed,	$aV\div6{,}500$		$0.33d$
Corliss,	$aV\div6{,}000$		$0.3d$
Exhaust-pipes, area:			
Slide-valve,	....	diam.=	$0.33d$
High speed,	$aV\div4{,}400$		$0.375d$
Corliss,	$aV\div3{,}800$		$0.33d$ to $0.37d$
Fly-wheel weight, in lbs. per H.P.:			
Slide-valve,	....		33
High speed,	$1{,}200{,}000{,}000{,}000\div d_1^2N^3$		25 to 33
Corliss,	....		80 to 120
Weight of engine:			lbs. per H.P.
Slide-valve,	....		125 to 135
High speed,	115 lbs. per I.H.P.		90 to 120
Corliss,	175 " " "		220 to 250

Piston speed in ft. per min. $=600$; weight of reciprocating parts in lbs., for high-speed engines $=1{,}860{,}000d^2\div lN^2$; square feet of belt surface per I.H.P. per min. $=55$ (high speed) and 35 (low speed) (Barr).

Clearance space: Corliss, $0.02l$ to $0.04l$; high speed, double valve, $0.03l$ to $0.05l$; high speed, single valve, $0.08l$ to $0.15l$; slide-valve, $0.06l$ to $0.08l$. Pressures on wearing surfaces in lbs. (L=length, D=diam., both in in.): Main bearings, $140LD$ to $160\ LD$; crank-pins, $1{,}000LD$ to $1{,}200LD$; cross-head pins, $1{,}200LD$ to $1{,}600LD$ (Stanwood).

Pressure on thrust-bearings $=35$ to 40 lbs. per sq. in. of area (Fowler).

Receiver volume for compound engine: If the cylinders are tandem, the connecting steam passages will be sufficient. If the cranks are at 90°, the volume of receiver should be at least as great as that of the low-pressure cylinder.

TEMPERATURE-ENTROPY DIAGRAMS.

In an indicator diagram the co-ordinates are pressure and volume and the area represents work done per stroke, in ft.-lbs.

In a temperature-entropy diagram the vertical ordinates are absolute temperatures, the horizontal ordinates, or abscissas, are quantities termed entropy, and the area represents energy measured in heat-units. Entropy, therefore, is length in a diagram whose area represents energy in heat-units and whose height is absolute temperature.

Isothermals on this diagram are horizontal straight lines,—the temperature being constant,—and adiabatics are vertical straight lines,—there being no change in the quantity of heat during a change of temperature.

Application to Carnot Cycle (Fig. 18). Heat supplied at τ_1 = area H_1, and heat rejected at τ_2 = area H_2, AB and CD being isothermals and BC and AD being adiabatics. Work done $= H_1 - H_2$, and efficiency $= (H_1 - H_2) \div H_1 = (\tau_1 - \tau_2) \div \tau_1$.

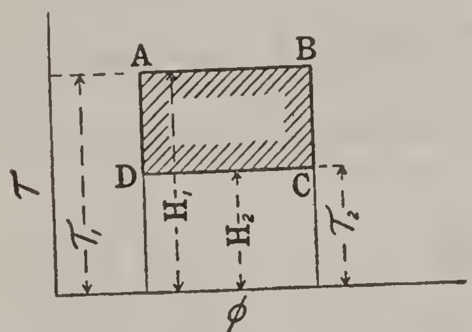
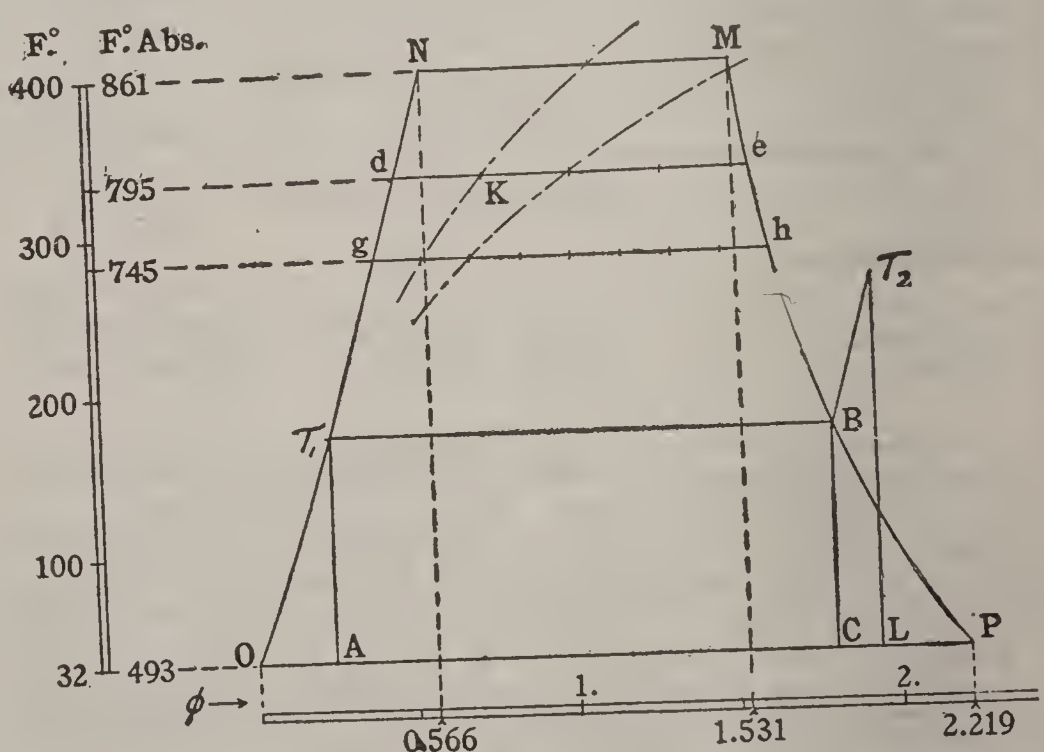


Fig. 18.

Construction of Diagram for Water and Steam. The diagram is drawn to represent the changes of 1 lb. of working substance and an arbitrary zero point is chosen to work from (i.e., 32° F. or 492° absolute). The entropy of water, then, at 492° = 0. At any other absolute temperature, τ, the entropy of water, $\phi w = \log_e \tau - \log_e 492 = \log_e \tau - 6.198$.

The additional entropy due to the conversion of water into steam is



g. 19.

equal to the latent heat (or heat necessary to convert the water into steam)

divided by the corresponding absolute temperature, or $L \div \tau = \phi_s$. The following table gives the

Entropy per Lb. Weight.

t	τ	Water from 32° F. (ϕ_w).	Steam (ϕ_s).	Steam and Water ($\phi_w + \phi_s$).
32	492	0.0000	2.2189	2.2189
50	510	.0359	2.1163	2.1522
100	560	.1296	1.8649	1.9945
150	610	.2154	1.6547	1.8701
200	660	.2949	1.476	1.7709
250	710	.3690	1.322	1.691
300	760	.4386	1.188	1.6266
350	810	.5042	1.0698	1.574
400	860	.5665	0.9649	1.5314

The results in this table are plotted in Fig. 19, ON being the water line or the plotting of the values of ϕ_w, and MP the dry-steam line, or $\phi_w + \phi_s$. If 1 lb. of water is raised from 32° F. to τ_1, the heat units required will be represented by the area $O\tau_1A$. The heat then required to convert the water into steam will be the area $\tau_1BCA\tau_1$ The entropy of the water will be OA as measured by the scale, that of the latent heat by AC, and the entropy of the steam and water by $OC (= OA + AC)$.

From steam-tables it is found that 1 lb. of dry saturated steam at 334° F. (794° ab.) occupies 4 cu. ft. If the isothermal at this temperature be divided into four equal parts, each part will represent 1 cubic foot. Also gh may be divided into eight parts, each representing 1 cu. ft. (1 lb. = 8 cu. ft. at 284° F.). Other isothermals may be similarly divided, and if all of the points for say 1 cu. ft are connected, the resulting curve will be a curve of constant volume (for 1 cu. ft.).

If 1 lb. of water at 334° F. be supplied with heat sufficient to evaporate one-quarter of itself, the distance dK will represent the portion of the total

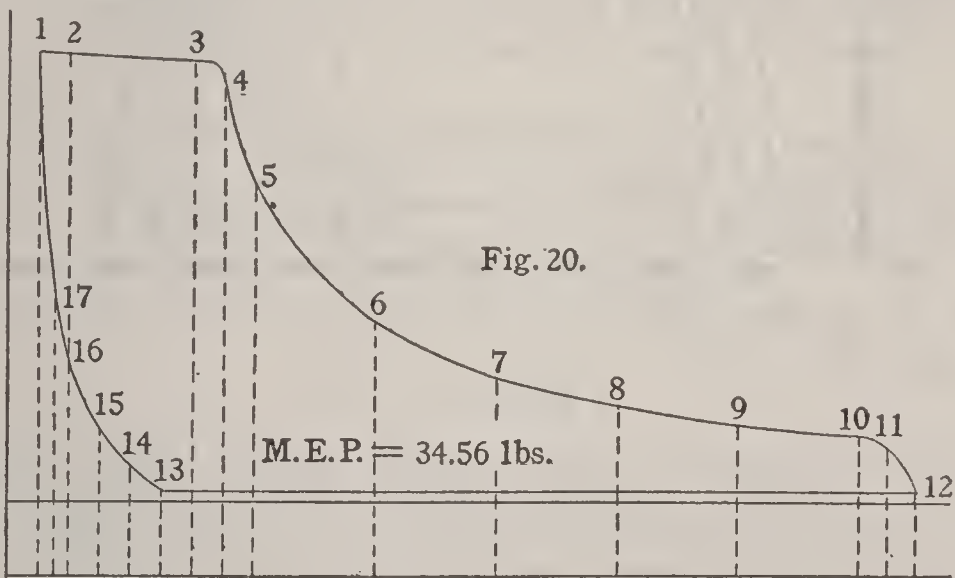


Fig. 20.

heat de required for the whole lb. The dryness of the steam (¼ of it being evaporated) will then be 0.25, and it may be stated that. The dryness is represented in the entropy diagram by the fraction (hor. dist. of point from water line) ÷ (hor dist. bet. steam and water lines) $= dK \div de$ in the instance under consideration.

If the steam is superheated to τ_2 before entering the cylinder, the additional entropy, CL, is obtained from the formula: Entropy, $CL = 0.48(\log_e \tau_2 - \log_e \tau_1)$.

To Draw the Entropy Diagram from the Data in an Indicator Diagram.—Fig. 20 is the indicator diagram of an engine having the following data: Initial pressure, 105 lbs., back-pressure, 17 lbs. (both absolute);

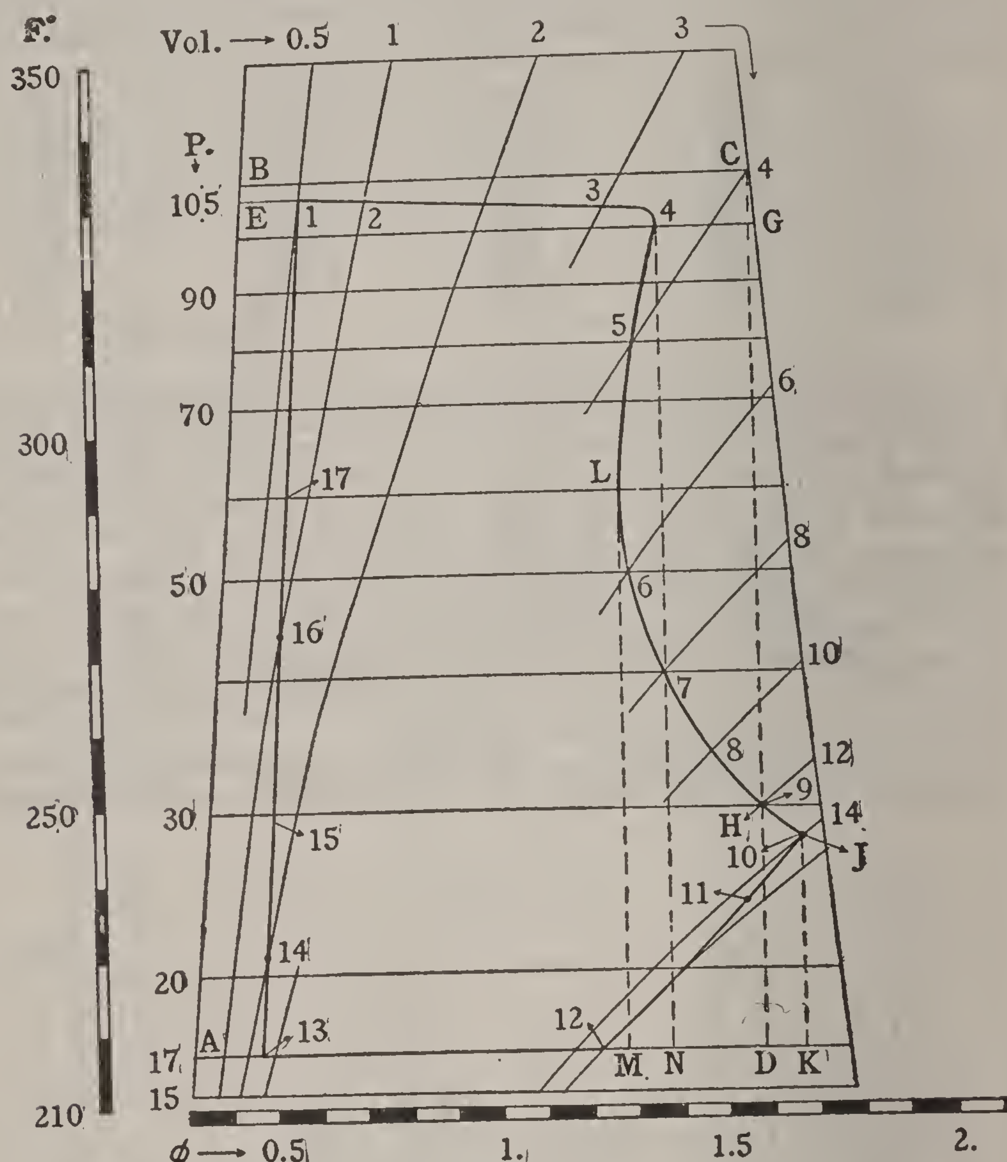


Fig. 21.

r.p.m. = 90; cylinder, 14×36; m.e.p. = 34.56 lbs.; I.H.P. = 87.06; area of cyl. = 153.94 sq. in.; volume of cyl. = 3.207 cu. ft.; volume of clearance (3.448%) = 0.11058 cu. ft.; lbs. steam used per hour = 2,133.5 (= 24.5 lbs. per I.H.P. hr.); lbs. of entering steam per stroke = 0.197547.

The compression steam is generally assumed to be dry, and, at point 17 (where vol. = 0.16587 cu. ft. and pressure = 60 lb.). its weight will be = 0.16587 × 0.14236 (or the weight of 1 cu. ft. at 60 lbs.) = 0.023613 lb. ∴ Total steam in cyl. = 0.197547 + 0.023613 = 0.22116 lb. and the vol of 1 lb. of steam similar to that in the cylinder, x = actual vol. in cyl. ÷ 0.22116. The pressures and values of x for the various points of Fig. 20 may now be plotted on Fig 21. For example, the pressure at point 7 on the indicator diagram is 40 lbs. (absolute). The contents of cyl. at this

point are 1.7694 cu. ft., which, divided by 0.22116, gives the volume x. of 1 lb., or 8 cu. ft. and point 7 on the entropy diagram is thus located by the intersection of the constant-volume curve 8 and the horizontal line of temperature 267° F. (727° abs.), which corresponds to a pressure of 40 lbs. absolute.

Losses. The entropy diagram just considered may be compared with that of the Rankine cycle for an ideal engine where the expansion is adiabatic down to back-pressure and where there is no compression. This latter diagram is the area *ABCDA*, *BC* being drawn at 108 lbs. (assuming a drop of 3 lbs. from the separator to cylinder).

The loss *BE4GCB* is that due to wire-drawing during the entrance of the steam; loss 4*GH*64 occurs during expansion and is due to condensation, leakage, etc.; loss *JK*12*J* is due to incomplete expansion; loss 13*AE*1 16 13 is due to clearance, compression, etc. All areas represent heat-units according to scale. The area 4*LMN*4 represents additional liquefaction loss after cut-off, and 7*NKJ*7 the gain due to re-evaporation. Fig. 21 shows only the working part of diagram, the full diagram on a smaller scale being shown by Fig. 21*a*.

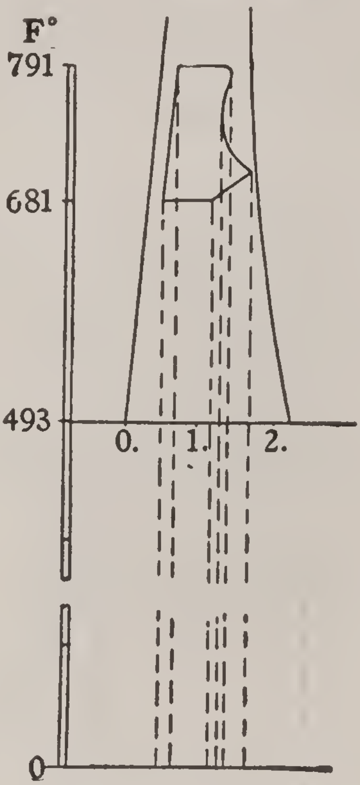


Fig. 21a.

Entropy Diagrams Applied to Internal Combustion Engines. $\phi = H \div \tau$; $d\phi = dH \div \tau$. $dH = k_v d\tau + (AP \div J)dV$, and $(AP \div J) = (k_p - k_v)\tau \div V$, or, combining these equations, $dH \div \tau = d\phi = (k_v d\tau \div \tau) + (k_p - k_v)dV \div V$, which is the general equation for change of entropy. (A = numerical constant, J = Joule's equivalent = 778, P = lbs. pressure per sq. ft.) Integrating between limits, $\phi_1 - \phi_2 = k_v \log_e (\tau_1 \div \tau_2)$ when the volume is constant, and $\phi_1 - \phi_2 = k_p \log_e (\tau_1 \div \tau_2)$ when the pressure is constant.

When P and V vary according to the law PV^x = constant, considering that $PV = R\tau$, letting $k_p \div k_v = \gamma$, substituting in the general equation and reducing, $\phi_1 - \phi_2 = k_v \dfrac{x - \gamma}{x - 1} \log_e \dfrac{\tau_1}{\tau_2}$, or, the change in entropy when PV^x = constant.

In adiabatic expansion $\gamma = x$, hence $\phi_1 - \phi_2 = 0$.

In the theoretical gas-engine diagram (Fig. 22, I.) $P_b = P_a V_a^{\gamma} \div V_b^{\gamma}$, and $\tau_b = P_b V_b \div (K_p - K_v)$, where V_b = specific volume of explosive mixture at b, K_p and K_v = specific heats of mixture in ft.-lbs. (= k_p and k_v multiplied by 778, or the equivalent of 1 heat-unit in ft.-lbs. In the following calculations the old value,—772,—has been employed). If τ_a is known, $\tau_b = \tau_a (r)^{\gamma - 1}$, where $r = V_a \div V_b$ and $\gamma = k_p \div k_v$. $\tau_c = \tau_b P_c \div P_b$.

The increase of entropy during the explosion is represented by the logarithmic curve bc (II, Fig. 22) and increase of entropy from b to c = $\phi_c - \phi_b = k_v \log_e (\tau_c \div \tau_b)$. Adiabatic expansion is shown by the vertical line cd, there being no change in the amount of entropy. $\tau_d = P_d V_d \div (K_p - K_v)$ and $P_d = P_c V_c^{\gamma} \div V_d^{\gamma} = P_c V_b^{\gamma} \div V_a^{\gamma}$.

From d to a (exhaust at const. vol.), $\phi_d - \phi_a = k_v \log_e (\tau_d \div \tau_a)$ which is negative. The exhaust and suction strokes do not enter into consideration, the temperature being assumed as constant.

The diagram is completed by drawing OX at the absolute zero of temperature, when the work done per cycle = area $abcd$; heat received per cycle = area $ObcX$; thermal efficiency = $abcd \div ObcX$; heat rejected into exhaust = area $OadX$.

Since $(\phi_c - \phi_b) = (\phi_d - \phi_a)$ and bc is governed by the same law as ad, the ratio of the two temperatures is constant and dependent only on the amount of compression, a high ratio resulting in a correspondingly increased efficiency.

The indicator card of a Crossley Otto engine tested by Prof. Capper

is shown in III, Fig. 22, the data for and a more complete analysis of which may be found in Golding's "Theta Phi Diagrams."

Cylinder, 8.5 in. diam. by 18 in. stroke, vol. = 0.591 cu. ft., clearance vol. = 0.2467 cu. ft., total vol. = 0.8377 cu. ft. R.p.m. = 162.5, explosions per min. = 71.2, net I.H.P. = 13.32. Gas used per hour = 279.75 cu. ft., gas per explosion = 0.06544 cu. ft. at 518° F. and 14.8 lbs. pressure, abso-

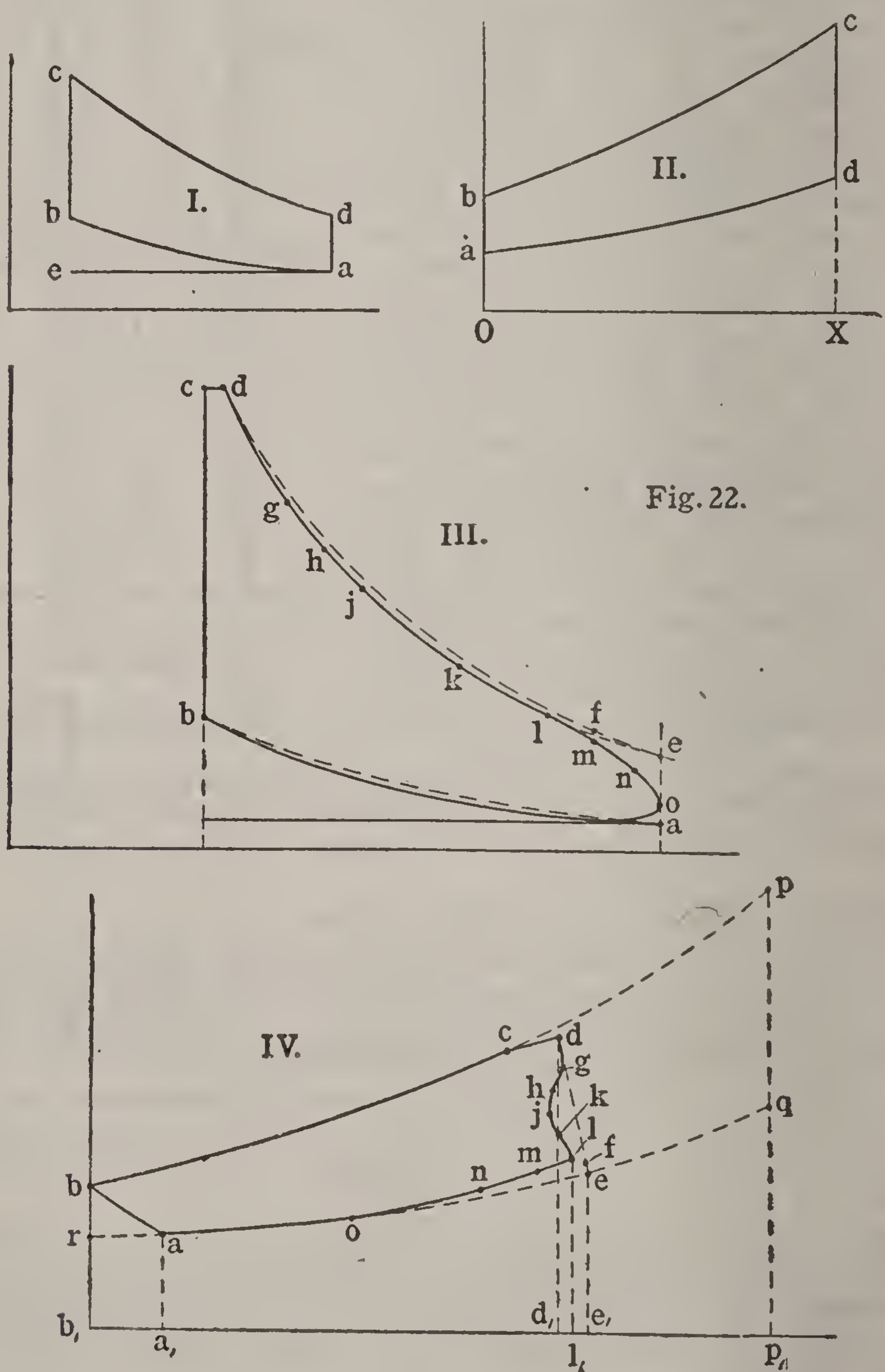


Fig. 22.

lute (= 0.0822 cu. ft. at temperature and pressure at a, or 605° and 13.8 lbs.) Pressures in lbs. per sq. in. at a, b, c, d and e = 13.8, 67.8, 240, 240 and 48.71, respectively. Volumes in cu. ft. at same points = 0.8377, 0.2467, 0.2467, 0.2617 and 0.8377, respectively. Since $p_a V_a^x = p_b V_b^x$, from the above values of p and V, $x = 1.3707$ for the ideal expansion curve = 1.3022 for the compression curve (both dotted). The location of e is found by

producing the actual expansion curve until it intersects the vertical *ae*. The coal gas (London) used had the following percentages by weight: CH_4, 42.79; C_2H_4 and C_4H_8, 18.21; H, 8.69; CO, 18.33; N, 7.14; CO_2 and O, 4.84. 1 cu. ft. = 0.0329 lb. $k_v = 0.5279$, $k_p = 0.6961$. The products of combustion or exhaust gases had the following composition (by weight): CO_2, 10.17; O, 6.7; N, 83.18. $kv = 0.1716$, $k_p = 0.2385$, 1 cu. ft. = 0.082 lb.

The clearance (filled with exhaust gases) held $0.2467 \times 0.082 = 0.02023$ lb. at 492° and 14.7 lbs., or, $[(0.02023 \times 4.2 \times 14.8) \div (605 \times 14.7)] = 0.01656$ lb. at 605° and 14.8 lbs. pressure at the beginning of suction stroke. The gas (0.06544 cu. ft.) having a specific volume of 34.87 cu. ft. per lb. at atmospheric pressure and temperature weighed 0.001877 lb. (Vol. at 605° and 14.8 lb. = 0.0822 cu. ft.) Air per explosion = $0.8377 - (0.2467 + 0.0822) = 0.5088$ cu. ft., which, at 605° and 13.8 lbs. pressure at *a* weighed 0.03131 lb. (16.25 cu. ft. per lb.). Total weight of mixture = 0.049747 lb.

Specific heats of mixture $K_v = 141.43$ ft.-lbs., $K_p = 199.09$ ft.-lbs., $K_p - K_v = 57.66$ ft.-lbs., $k_p = 0.25788$, $k_v = 0.1832$, $\gamma = 1.4077$. From these values and the previously given temperature equations, $\tau b = 840°$ F. (absolute), $\tau_c = 2{,}973°$, $\tau d = 3{,}154°$, $\tau e = 2{,}048°$, and $\tau_a = 580°$. (This is 25° lower than the value assumed, 605°, but the difference need not be considered.)

Taking entropy at *b* as zero, the entropy at $c = \phi_c - \phi b = kv \log_e (\tau_c \div \tau b) = 0.23158$. $\phi d - \phi_c = k_p \log_e (\tau d \div \tau_c) = 0.25788 \times \log_e(3{,}154 \div 2{,}973) = 0.01524$. $\phi e - \phi d = kv\dfrac{x-\gamma}{x-1} \log_e(\tau d \div \tau e) = 0.1832[(1.3707 - 1.4077) \div 0.3707] \log_e(3{,}154 \div 2{,}048) = 0.00709$. $\phi_a - \phi_e = kv \log_e (\tau e \div \tau_a) = -0.23112$. $\phi b - \phi_a = kv\dfrac{x-\gamma}{x-1} \log_e (\tau b \div \tau_a) = -0.02369$. Positive entropy, *b* to $e = 0.23158 + 0.01524 + 0.00709 = 0.25472$. Negative entropy, *e* to $b = 0.23112 + 0.02369 = 0.25481$. The two sums should exactly balance, the slight difference being due to insufficiently extended calculations.

The diagram for the ideal cycle is represented by *abcdea* (IV, Fig. 22), whose area = 171.875 B.T.U., or the work performed by 1 lb. of the mixture. The work per explosion (i.e., of 0.049747 lb.) = 8.55 B.T.U. = 6,600 ft.-lbs. The actual cycle is now to be considered. The curves *bc* and *cd* in the entropy diagram are correct, but during expansion the actual curve of pressures differs considerably from the ideal or dotted curve, and it is therefore necessary to select several points on the actual curve and calculate the temperature and entropy at each. These values are given in the following table:

Point on Diagram.	P, Lbs. per Sq. In.	V, in Cu. Ft.	τ, in Degs. F.	x.	Entropy (ϕ).
d	240	.2617	3,154		.24682
g	170	.335	2,858	1.3965	.24734
h	134	.394	2,650	1.4668	.24559
j	109	.453	2,478	1.4798	.24374
k	80.5	.572	2,312	1.2995	.24837
l	62.5	.6897	2,164	1.3526	.25027
m	53.8	.749	2,023		.24404
n	38	.808	1,541		.200
o	24	.8377	1,009		.1217

At *m*, just after release, $p = 53.8$ lb.; the pressure at *f* on ideal curve (vertically above *m*,—at same vol.) = 56.79 lb.; $\tau m = Pm\,Vm \div (K_p - K_v) \times 0.049747 = (53.8 \times 144) \times 0.749 \div (57.66 \times 0.049747) = 2{,}023°$; $\tau f = 2{,}135°$. The drop of entropy from 2,135° to 2,023° = $0.1832 \log_e (2{,}135 \div 2{,}023) = 0.00991$, which must be subtracted from the entropy at *f*.

The entropy at *f* in excess of that at $d = kv\dfrac{x-\gamma}{x-1}\log_e(\tau d \div \tau f) = 0.00713$, consequently the entropy at *m* = entropy at *d* + additional entropy to *f* − drop in entropy from *f* to $m = 0.24682 + 0.00713 - 0.00991 = 0.24404$. Values for points *n* and *o* are similarly obtained, the results being included in above table.

The heat transformed into work = area *abcdjloa*. This, however, does not represent the total heat generated during the explosion. The total available heat of each explosion = 36.04 B.T.U. (or that of 0.001877 lb. of gas, whose calorific value is 19,200 B.T.U. per lb.). To represent this on the diagram, produce *bc* to *p* so that the area $b_1bpp_1 = 36.04 \div 0.049747 =$ 724.5 B.T.U. per lb. of mixture. $\tau_p = \tau b + \tau_r$ (τ_r = the rise in temperature from *b* due to complete combustion). $\tau_r = 724.5 \div kv = 3{,}955°$ and $\tau p =$ $3{,}955 + 840 = 4{,}795°$. Net heat transformed into work = *abcdjlo* = 8.2 B.T.U. per explosion, or 22.75% of the total available heat. Heat given to cylinder walls during compression stroke = $a_1abb_1 = 0.77$ B.T.U Heat given to exhaust = $a_1aomll_1 = 13.63$ B.T.U The remainder ($l_1ljdcpp_1$ = 13.44 B.T.U.) is transmitted through the cylinder walls, and the total heat passing through walls = 13.44 + 0.77 = 14.21 B.T.U = heat given to jacket water plus that radiated from the exterior surface of cylinder head and piston.

In an ideal engine (i.e., one with a non-conducting cylinder, complete combustion, exhaust at constant volume, adiabatic expansion and compression) the work per explosion = area *rbpq*, and the maximum possible work = 100 $(\tau b - \tau_a) \div \tau b$ per cent of the total heat evolved, $= 100(840 - 580) \div 840 = 30.95\%$ of the 36.04 B.T.U. = 11.154 B.T.U. per explosion. The net work actually obtained = 8.2 B.T.U. = 73.5% of the maximum. The same general method is employed for oil engines, temperatures being calculated from $PV = R\tau$, etc. In a Diesel engine where oil is sprayed into the cylinder under air pressure for 5 to 10% of the combustion stroke, $k_p = 0.264$ (mean value) and if γ is taken at 1.408, $kv = 0.1875$.

STEAM TURBINES.

Turbines are machines in which a rotary motion is obtained by means of the gradual change of the momentum of a fluid.

In steam turbines the energy given out by steam during its expansion from admission to exhaust pressure is transformed into mechanical work, either by means of pressure or of the velocity of the steam while expanding.

The De Laval turbine is one of pure impact and consists of a wheel carrying a row of radially attached vanes or buckets. The steam is delivered to these vanes from stationary nozzles, in which it is fully expanded (thus attaining the highest practicable velocity) and after passing the vanes is exhausted either into the atmosphere or into a condenser. The nozzles are inclined to the plane of the wheel at an angle of 20°; the inlet and outlet angles of the vanes range from 32° to 36° according to the size of the turbine. The best peripheral velocity is about 47% of the steam velocity. Economical reasons restrict it to about 1,400 ft. per sec. for large wheels and 500 ft. per sec. for small ones. R.p.m. of wheels range from 10,000 to 30,000, and are reduced to 0.1 these values by helical gears.

In the Parsons turbine a drum with rows of radial vanes revolves in a stationary case. Between each row of moving vanes there is a ring of vanes fixed to the case which deflects the direction of the steam flow to the next rotating row of vanes. The diameters of drum and casing increase in steps from inlet to exhaust end, the steam flowing through the vanes in the annular space between the drum and case. The expansion is practically adiabatic

The Rateau multicellular turbine in effect consists of a number of wheels of the De Laval type mounted side by side on the same shaft, each wheel rotating in a compartment of its own and the exhaust of each wheel being led through nozzles or openings in the partition walls to the next succeeding wheel. Step-by-step expansion and moderate speeds are thereby obtained.

In the Curtis turbine the nozzles deliver steam at a velocity of about 2,000 ft. per sec. and this velocity is absorbed by a series of moving vane wheels on a vertical shaft with alternating fixed rings of stationary guide blades, similar to Parsons' arrangement.

When the initial velocity has been absorbed the steam is again expanded through another set of nozzles to a further series of wheels, and so on. By this compounding the peripheral speed is kept down around 400 ft. per sec. In the following table the pressures are gauge pressures.

Steam Turbine Data.

Make.	Size.		Vacuum.	Steam Pressure.	R.P.M.	Superheat.	Lbs. Steam per Hour 0.5 Load.	Full Load.		
			in.	lb.		° F.				
Parsons.	400	K.W.	25	125	3,300	0	.....	15.41	per	B.H.P.
'	1,250	"	25	150	1,200	.....	.....	14.4	"	"
"	1,250	"	28	150	1,200	77	.....	13.2	"	E.H.P.
De Laval.	30	H.P	....	100	2,000	.....	41	40	"	B.H.P
"	30	"	....	50	2,000	.....	50	50	"	"
"	30	"	25.5	125	2,000	.....	25–30	22	"	"
"	300	"	27	200	900	20–90	16.5	14.5	"	"
"	300	"	27	200	900	.....	17.5	15.5	"	"
Curtis.	2,000	K.W	28.8	160	750	242	16.3	15.3	"	K.W.
Rateau	500	H P	(1.33 abs.)	62	2,400	.....	18		"	E.H.P.
"	500	"	(1.63 abs.)	121	2,400	.....	.....	15.8	"	"
"	500	"	29	180	2,400	90	.....	11.5	"	"
Westinghouse-Parsons.	600	"	28	150	.....	100	14.34	12.48	"	B.H.P.
"	600	"	28	150	.....	0	15.86	13.89	"	"

Flow of Steam through Nozzles. Zeuner's formula for the velocity of steam flowing through a nozzle and expanding adiabatically may be simplified to the following form without involving appreciable error: v (in ft. per sec.) $=224\sqrt{h-h_1+ls-l_1s_1}$ (1), where h and h_1 are the initial and final heat in the water in B.T.U., l and l_1 the initial and final latent heat in the steam in B.T.U., and s and s_1 are the initial and final degrees of saturation of the steam.

$s_1=s-(t-t_1)(c-t)x\cdot 10^{-7}$ (2), where s_1=saturation after adiabatic expansion, s=initial saturation, t and t_1 are temperatures (F °) before and after expansion.

Values of c and x. (s is assumed or ascertained beforehand.)

When $s=$	1	.95	.90	.85	.80	.75	.70
$c=$	900	870	845	833	817	770	710
$x=$	16.6	15.7	14.7	13.4	12	11.5	11

The weight of steam delivered per sq. in. of nozzle cross-section per minute in lbs., $w=0.417v\div su$ (3), where u=cu. ft. in 1 lb. of dry steam at the pressure corresponding to v.

At that section of the nozzle where the pressure has dropped to 58% of the initial pressure the flow per sq. in. is greatest, hence this section is the smallest and the nozzle diverges from this point to the mouth.

The theoretical minimum weight of steam per H.P. hour, $W=127{,}000{,}000\div v^2$ (at mouth) (4).

(The foregoing matter has been derived from an article by A. M. Levin in Am. Mach., 6–30–04.)

Example —Steam at 185 lbs. (absolute) containing 20% of moisture ($s=0.8$) is required to expand adiabatically in a nozzle to 1 lb. (absolute). p at throat $=185\times0.58=107.3$ lbs. From formula (2) and steam-tables the following values are found

p.	lbs.	$t°$.	l.	s.	u.	h.
Initial.......	185	375	848	0.800	2.45	348
Throat.......	107.3	333	879	.778	4.08	304
Mouth.......	1	102	1,043	.655	334	70

Substituting in (1) and (3), v at throat $=1{,}391$ ft. per sec., v at mouth $=3{,}703$ ft. per sec., w at throat $=182.75$ lbs. per sq. in. per min., and w at mouth $=7.058$ lbs. per sq. in. per min.

Area of cross-section at mouth $=(182.75\div7.058=25.9)\times$section at throat. Min. wt. of steam per H.P. hour (from (4)) $=9.27$ lbs. The kinetic energy of 1 lb. steam $=v^2\div2g$; if $v=3{,}703$, kinetic energy $=213{,}200$ ft.-lbs.

In designing a nozzle, calculate v at mouth from the conditions assumed — then v^2(mouth)$\div2g$=kinetic energy of 1 lb. of steam in ft.-lbs. Assume

this energy to develop from 0 at the inlet to its full value at the mouth by equal increments per increment of nozzle length, and plot curve of velocities corresponding thereto. Assume several pressures between supply and mouth and find the corresponding velocities from (1), locating these pressures vertically under the corresponding velocities on the curve, and draw a second or pressure-curve through these points. Determine s, h, l, and u from steam-tables and formula (2) and find values of w by formula (3) for the various pressures chosen. The reciprocals of w will be the sq. in. of cross-section per lb. of steam per min., which, if plotted, will give points in the curve of nozzle cross-section.

(For an elaboration of this subject, consult Stodola's "The Steam Turbine," translated by Dr. L. C. Loewenstein, D. Van Nostrand Co.)

LOCOMOTIVES.

Train Resistance. $R_1=3\left(\frac{V+12}{V+3}\right)+\frac{V^2}{300}$ (European practice, Fowler's Pocket Book); $R_1=3+\frac{V}{6}$ (Baldwin Loco. Wks.); $R_1 = 4 + 0.005V^2 + (0.28+0.03N)\frac{V^2}{W}$ (Wellington); $R_1=4+\frac{V^2}{130}$ (Wellington, for any loading, 5 to 35 mi. per hr.); $R_1=3+.0386V+\left(\frac{17.1}{w}+1.036\right)\frac{V^2}{1,000}$ (Von Borries). In these formulas R_1 = resistance in lbs. per ton of 2,000 lbs. (2,240 lbs. for first formula), V = speed in miles per hour, N = number of cars in train, W = weight of train in tons of 2,000 lbs., and w = wt. of one car in tons.

Resistance due to grade in lbs. per ton (2,000 lbs.), $R_2=0.3788G$, where G = grade in feet per mile.

Curve resistance, in lbs. per ton, $R_3=0.5682A$, where A = angle of curve in degrees. (The angle of a railway curve is the angle at the center subtended by a chord of 100 ft. The radius of a curve of A degree = 5,729.65 ft. $\div A$.)

Acceleration resistance (due to change of speed), $R_4=0.0132(V_1^2-V^2)$, where V_1 is the higher speed.

Total resistance, $R=R_1+R_2+R_3+R_4$.

Horse-Power $=(WVR\times5{,}280)\div(33{,}000\times60)=0.002666WVR$.

Tractive Power cannot exceed the adhesion, which varies from 20% of the weight on the drivers when rails are wet or frosty, to 22.5% when dry. At starting 25% may be attained by the use of sand.

Tractive power $=d^2p_1s\div d_1$, where d and d_1 are respectively the diams. of cylinder and drivers in in., p_1 the mean effective pressure in lbs. per sq. in., and s = stroke in in. M.E.P. = boiler pressure $p\times c$ (approx.). Values of c:

Cut-off = . .	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1
c = ...	0.2	0.4	0.55	0.67	0.79	0.89	0.98	1

The average m.e.p. decreases as the piston speed increases, as shown in the following from Bulletin No. 1, Am. Ry. Eng. & Maintenance of Way Assn.:

Piston speed (ft. per min.).	250	300	400	500	600	800	1,000	1,200
M.E.P. (%).	85	80.2	70.8	62	54	40.7	31.6	26

For compound engines of the Vauclain 4-cyl. type, Tractive power in lbs. $=ps(2.66D^2+d^2)\div4d_1$, where p = boiler pressure, and D = diam. of high-pressure cyl. (For a 2-cyl. or cross-compound, omit d^2 from formula.)

The tractive power decreases as the speed increases, as shown by the following table, where r = stroke $\div$ diam. of driver, and a speed of 10 mi. per hr. is taken as unity.

V = ...	10	15	20	25	30
(r=0.429). .	1	.88	.75	.64	.53
(r=0.536). .	1	.83	.67	.54	.45

Weight of Train in tons, for average freight work (including engine and tender) W = tractive power $\div [6+20\times$(grade in per cent)]. The weight of freight carried may be taken as (W − wt. of loco.) $\div 2$. H.P. = Tractive power $\times V \div 375$.

Grate Area in sq. ft. $= d^2s \div C$ (d and s in in.). For express locomotives, simple, $C = 197$ to 288 (average practice = 240); compound, $C = 118$. For freight locomotives, simple, $C = 250$ to 290 (= 500 for very heavy locos.); compound, $C = 132$ to 197 (= 177 for good practice). (For compound locos. d = diam. of h. p. cyl.)

Heating Surface = Grate area $\times C$. For passenger locomotives, C = 47 to 75 (= 70 for good practice). For freight locos., C = 65 to 100 (best practice on heavy locos., C = 78 to 90).

Diameters of Cylinders. $d = 0.542\sqrt{d_1 w \div ps}$, where w = weight on drivers in lbs. For the diam. of h. p. cyl. in a compound engine replace 0.542 in formula by 0.4 to 0.46. Diam. of l. p. cyl. = (1.56 to 1.72) $\times$ diam. h. p. cyl.

Areas of Steam-Ports. For simple locos., $A = 7.5\%$ of cyl. area. For heavy, modern freight locos., $A = 10\%$ of area of h. p. cyl. and 4.5 to 6.5% of l. p. cyl. area.

Areas of Exhaust-Ports, simple, about $2.5\times$ area of steam-port.

Piston-Valves. Diam. of valve $= 0.4\times$ cyl. diam.

Coal Consumption. From 120 to 200 lbs. per hour per sq. ft. of grate area.

Under favorable conditions one I.H.P. requires the combustion of 4 to 5 lbs. of coal per hour.

Balancing. To avoid oscillations the forces and couples in a horizontal plane due to the inertia of the reciprocating parts must be eliminated as far as possible.

Let W = combined weight of crank-pin, connecting-rod, cross-head, piston-rod and piston + one-half the weight of one crank-arm. (In the case of an inside cyl. take the weight of one web in place of $\frac{\text{crank-arm wt.}}{2}$); r = radius of crank; R = radius of c. of g. of balance-weight; a = distance between centers of wheels (i.e., c. to c. of rails); b = distance between centers of cyls. Then, the weight of each balance-weight, $W_b = \frac{Wr}{aR}\sqrt{\frac{a^2+b^2}{2}}$, and $\tan\theta = \frac{a-b}{a+b}$, where θ = angle between radius to c. of g. of balance-weight from wheel center, and the center line of the near crank produced. For inside cyls. both balance-weights fall within the quadrant bounded by the produced center lines of the cranks. For outside cyls. $\tan\theta$ is negative and the balance-weights are outside of the said quadrant.

In the U. S. the balance-weights are equally divided between the wheels coupled together; in England they are concentrated on the drivers. The U. S. method reduces the hammer-blow on the rails, and to still further lessen this, some builders balance only 75% of the reciprocating weight.

Another rule is as follows: On the main drivers place a weight equal to one-half the weight of the back end of the connecting-rod plus one-half the weight of the front end of connecting-rod, piston, piston-rod, and cross-head. On the coupled wheels place a weight equal to one-half the weight of the parallel-rod plus one-half the weights of the front end of the main-rod, piston, piston-rod, and cross-head. Balance-weights to be opposite the crank-pins and their centers of gravity must be at the same distances from the axles as the crank-pins.

Friction of Locomotives. An 8-wheel Schenectady passenger locomotive tested by Prof. W. F. M. Goss gave the following results. (Cyl. 17×24, drivers, 63 in., wt., 85,000 lbs.)

Cut-off at $\frac{1}{4}$ stroke,	friction at	15 mi. per hr.	=	12%	of total power.
" " $\frac{1}{4}$ "	" "	55 " " "	=	23%	" " "
" " $\frac{1}{3}$ "	" "	15 " " "	=	7.4%	" " "
" " $\frac{1}{3}$ "	" "	55 " " "	=	15.7%	" " "

Dimensions of Modern Locomotives.

Type	Passenger				Freight					
Railroad	Penna.	A., T.&S.F.	Alton.	Prussian State. (Von Borries.)	B. & O. (Mallet Duplex Comp.)	N.Y. Cent.	Paris & Orleans. De Glehn, 4 cyl.	Atch., Top. & Santa Fé		
Builder	Am. Loco. Co.	Baldwin.	Baldwin.	Hanover Mch. Co.	Am. Loco. Co.	Am. Loco. Co.	Soc. Alsacienne.	Am. Loco. Co.	Baldwin.	Baldwin.
Arrangement	†4–4–2	4–4–2	4–6–2	4–4–2	0–12–0	2–8–0	4–4–2	2–10–0	2–8–2	2–10–0
Drivers, diam., in.	80	73	80	$77\frac{15}{16}$	56	51	80	57	57	57
Wt. on drivers, lbs.	109,000	90,000	141,700	67,000	334,500	201,000	79,500	232,000	199,670	237,800
" total, lbs.	176,600	187,000	219,000	132,650	334,500	227,000	161,200	259,800	261,720	267,800
" per driv'g-axle.	54,500	45,000	47,233	33,500	55,750	50,250	39,750	46,400	50,000	47,560
Cyls., diam., in.	20.5	15 & 25	22	$14\frac{3}{16}$ & $22\frac{1}{16}$	20 & 32	16 & 30	$14\frac{3}{16}$ & $25\frac{5}{8}$	$17\frac{1}{2}$ & 30	18 & 30	19 & 32
" stroke, in.	26	26	28	$23\frac{5}{8}$	32	30	$25\frac{3}{16}$	32	32	32
Boiler, type	Belpaire	Wagon top	Straight top	Straight top	Straight top	Ext. wagon top	Straight top	Ext. wagon top	Wagon top	Wagon top
" diam., in.	67	66	70	$56\frac{11}{16}$ inside	82 inside	77	$65\frac{5}{8}$ inside	$78\frac{3}{4}$	$78\frac{3}{4}$	$78\frac{3}{4}$
" working pres., lbs.	205	220	220	200	235	210	228	225	225	225
Tubes, diam & lgth	2″×15′ 1″	$2\frac{1}{4}$″×18′ 1″	$2\frac{1}{4}$″×20′	2″×14′ $7\frac{3}{16}$″	$2\frac{1}{4}$″×21′	2″×14′ 9″	$2\frac{3}{4}$″×14′ $5\frac{1}{4}$″	$2\frac{1}{4}$″×18′ 6″	$2\frac{1}{4}$″×19″	$2\frac{1}{4}$″×19′
" No.	315	273	328	241	436	507	*126 (Serve)	413	463	463
Heating surface, tubes, sq. ft.	2,474	2,839	3,848	1,632	5,366	3,915	2,403	4,476	5,156	5,156
Heating surface, fire-box, sq. ft.	165	190	230	108	219	227	174	206	210	234
Total H't'g surface.	2,639	3,029	4,078	1,740	5,585	4,142	2,577	4,682	5,366	5,390
Grate area, sq. ft.	55.5	49.4	54	29	72	58	33.4	59.5	58.5	58.5
Simple or compound	Simple	Bal. comp.	Simple	4 cyl bal. comp.	Tandem comp.	Tandem comp.	Bal. comp.	Tandem comp.	Vauclain comp.	Tandem comp.
Wheel base, drivers	7′ 5″	6′ 4″	13′ 9″	6′ $10\frac{11}{16}$″	30′ 8″	15′	7′ $\frac{5}{8}$″	20′	16′	20′ 4″
" " total	30′ $9\frac{1}{2}$″	25′	32′ 8″	29′ $6\frac{5}{16}$″	30′ 8″	23′ 7″	28′ $6\frac{1}{2}$″	28′ 11″	31′ $6\frac{1}{2}$″	29′ 10″
Superheater surface in sq. ft.	....	....	....	589						

* Serve tubes have internal longitudinal ribs which render cleaning difficult, but on account of increased surface they have from 30 to 50% greater efficiency.

† 4–4–2 indicates wheel arrangement, thus: 4 front truck wheels,—4 drivers,—2 trailers.

STEAM-BOILERS.

Horse-Power. The capacity of a boiler is fully expressed by stating the quantity of water it is capable of evaporating in a given time under given conditions, and the H.P. of the steam so generated depends entirely on the economy of the engine in which it is used. There is, however, a commercial demand for rating boilers in terms of H.P. and the A.S.M.E. committee has recommended the following: The unit of commercial H.P. developed by a boiler shall be 34.5 lbs. of water evaporated per hour from a feed-water temperature of 212° F. into dry steam of the same temperature, -hich is equivalent to 33,317 B.T.U. per hour and also practically equivalent to an evaporation of 30 lbs. of water from 100° F. into steam at 70 lbs. gauge pressure.

Heating Surface is all that surface which is surrounded on one side by water to be heated and on the other by flame or heated gases. Heating surface in sq. ft., $A = cQ \div H$, where Q = quantity of water evaporated per hour, H = total heat of the steam at boiler pressure, and c for locomotive boilers = 90, for Scotch marine boilers = 180, for Cornish = 220, for plain cylinder = 280, for return-tubular and water-tube boilers = 400.

Relative Values of Heating Surfaces per sq. ft. compared with flat plates Flat plate above fire, 1; cylindrical surface above and concave to fire, 0.95; same, but eonvex, 0.9; flat surface at right angles to the current of hot gases, 0.8; water-tube surface, same as last, 0.7; sloping surface at side of and inclined to the fire, 0.65; vertical surface at side of fire, 0.5; locomotive boiler tubes,—not more than 3 ft. from fire-box tube plate, 0.3. Horizontal surfaces underneath the fire and the lower half of internally heated tubes are not considered as effective.

Ratio of Heating Surface to Grate Surface. Plain cylinder, 10 to 15; Scotch marine and Cornish, 25 to 40; Lancashire, 26 to 33; horizontal return-tubular, 30 to 50; water-tube, 35 to 65; locomotive, 60 to 90.

Areas of Tubes and Gas Passages. Area near bridge wall = $\frac{1}{7}$ grate area. Tube area (total) = 0.1 to 0.11 × grate surface for anthracite and 0.14 to 0.17 × grate area for bituminous coal, both at moderate rates of combustion (Barrus).

Holding Power of Tubes. Expanded only, 5,000 to 6,000 lbs.; expanded and flared, 19,000 to 20,000 lbs.

Boiler Efficiencies. For the purpose of comparison it is customary to express the evaporation in lbs. of dry steam per lb. of pure eombustible, and in order to eliminate the effects of variation in the temperature of the feed-water, the results are reduced to what is termed "the equivalent evaporation" from and at 212° F. (See page 59.) The complete combustion of 1 lb. of pure carbon will evaporate $\frac{14,600}{965.7} = 15.3$ lbs. of water from and at 212°. 192 American boiler tests summarized by H. H. Suplee give 10.86 lbs. per lb. of fuel, which may be considered as good practice, ordinary averages being from 6 to 8 lbs. per lb. of fuel. 12.5 lbs. evaporation is generally the best obtainable from high-grade fuels like Pocahontas and Cumberland coals. One test, however, is recorded showing an evaporation of 13.23 lbs. per lb. of Cumberland coal.

Performance of Boilers (D. K. Clark). $w = Ar^2 + Bc$, where w = lbs. water evaporated from and at 212° F. per sq. ft. of grate per hour, r = ratio of heating to grate surface, and c = lbs. fuel per sq. ft. of grate per hour. A and B are respectively as follows: Stationary boilers, 0.0222 and 9.56; marine, 0.016 and 10.25; portable, 0.008 and 8.6; locomotive, 0.009 and 9.7

Materials and Tests. (From Am. Boiler Mfrs. Assn. Uniform Specifications.)

Cast Iron. Should be soft, gray, and highly ductile; used only for hand-hole plates, man-heads, and yokes.

Steel. Homogeneous open-hearth or crucible.

Shell Plates not exposed to direct heat. Tensile Strength (T.S.) 65,000 to 70,000 lbs. per sq. in.; elongation > 24% in 8 in. Phosphorus (P) and Sulphur (S) < 0.035%.

Shell Plates exposed to direct heat. T.S. = 60,000 to 65,000 lbs., elongation > 27% in 8 in., P < 0.03% and S < 0.025%.

Fire-Box Plates (exposed to direct heat). T.S. = 55,000 to 62,000 lbs., elongation > 30% in 8 in., P < 0.03% and S < 0.025%.

Test Pieces to be 8 in. long with a cross-section > 0.5 sq. in.; width = or > thickness, edges machined. Up to 0.5 in. thickness, plate must stand bending double and being hammered down flat upon itself. Above 0.5 in. it must stand bending 180° around a mandrel of diam. = $1.5t$. Bending-test pieces must not be less than $16t$ in length, edges must be machined and pieces must be cut both lengthwise and crosswise from plate.

Rivets must be of good charcoal iron or of soft mild steel having same properties as fire-box plates. They must be tested hot and cold by driving down on an anvil with the head in a die, by nicking and bending and by bending back on themselves cold, all without developing cracks or flaws.

Tubes to be of charcoal iron or mild steel made for this purpose, lap-welded or drawn. Tubes must be round, straight, free from blisters, scales, and other defects and tested under an internal hydrostatic pressure of 500 lbs. per sq. in. Standard thicknesses (B.W.G.):—No. 13 for 1 to $1\frac{3}{4}$ in. tubes, No. 12 for 2 to $2\frac{1}{2}$ in., No. 11 for $2\frac{3}{4}$ to $3\frac{1}{4}$ in., No., 10 for $3\frac{3}{4}$ and 4 in., No. 9 for $4\frac{1}{2}$ and 5 in.

Tube Tests. A section cut from one tube selected at random from a lot of 150 or less must stand hammering down vertically when cold without cracking or splitting. Tubes must also stand expanding flange over on tube plate.

For tubes...........	1 to $1\frac{3}{4}$	2 to $2\frac{1}{2}$	$2\frac{3}{4}$ to $3\frac{1}{4}$	$3\frac{1}{2}$ to 4	$4\frac{1}{2}$ to 5 in. in diam.
Length of test piece =	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$ in.

Stay Bolts of iron or mild steel must show on an 8 in. test piece as follows: Iron, T.S. > 46,000 lbs., elastic limit > 26,000 lbs., elongation > 22% for sections under 1 sq. in. and > 20% for larger sections.

For steel these values are respectively > 55,000 lbs., > 33,000 lbs., > 25%, and > 22%.

Tests. A bar taken at random from a lot of 1,000 lbs. or less and threaded with a sharp die to a V thread with rounded edges must bend cold 180° around a bar of same diam. without developing cracks or flaws. Another bar, screwed into a well-fitting nut of the material to be stayed and riveted over, must be pulled in a testing machine. If it fails by pulling apart its strength is measured by the T.S. If failure is due to shearing, the measure of strength is the shear stress per sq. in. of mean section in shear.

(Mean section $= \dfrac{t \text{ of plate}}{2} \times$ circumf. at half height of thread.)

Braces and Stays to be of same material as stay bolts. T.S. to be determined from a 10 in. bar from each lot of 1,000 lbs. or less.

All bending and hammering tests indicated above must develop no flaws, cracks, splitting, opening of welds, or any other form of distress.

Workmanship and Dimensions. Flanging, bending, and forming should be done at suitable heats, no bending or hammering, however, being allowed on any plate which is not red by daylight at the point worked upon and at least 4 in. beyond it. Rolling to be by gradual increments from the flat plate to a true cylindrical surface, including the lap. The thickness of bumped or spherically dished heads should equal that of a cylindrical shell of solid plate whose diam. is equal to the radius of curvature of the dished head, an increase of t being taken to allow for rivet holes, manholes, etc.

Rivet holes should be perfectly true and fair, either drilled or cleanly punched, burrs and sharp edges to be removed by slight countersinking and burr-reaming both before and after sheets are joined. Under sides of original rivet heads to be flat, square, and smooth. Allow length of $1\frac{1}{2}$ diam. for stock for heads, for $\frac{5}{8}$ to $\frac{13}{16}$ in. rivets, and less for larger sizes. Allow 5% more stock for driven head for button-set or snap rivets. For machine-riveting, total pressure on die = 35 tons for $\frac{3}{4}$ in. rivets, 57 tons for $\frac{15}{16}$ in. rivets, 65 tons for 1 in., and 80 tons for $1\frac{1}{8}$ and $1\frac{1}{4}$ in. rivets. Approximately, make d of rivet hole $= 2t$ (of thinnest plate), $p'' = 3d$, distance between pitch lines of staggered rows $= 0.5p''$, lap for single-riveting $= p''$, lap for double-riveting $= 1.333p''$ (add $0.5p''$ for each additional row of rivets). For exact dimensions make resistance to shear of aggregate rivet section = > 1.1 × T.S. of net metal. Holes < $\frac{5}{8}$ in. in steel may be punched, above $\frac{5}{8}$, punch and ream, or drill. Drift-pins to be used only to pull plates into position,—never to enlarge holes. Calking to be done only

with round-nose tools, calking edges to be planed, sheared, or chipped to a bevel. Finishing may be done with a square-nose tool if care is taken to avoid nicking the lower plate. Safe working pressure per sq. in. on flat surfaces $p=Ct^2 \div (p'')^2$, where t=thickness of plate in 16ths of an inch, p''=pitch of stays in in., and $C=112$ for plates $\frac{7}{16}$ in. and less, with riveted screw stays, 120 for plates $>\frac{7}{16}$ in. with riveted screw stays, and 140 for all plates where the screw stays have in addition a nut inside and outside the plate. This latter is imperative when the feed-water contains salt, acids, or alkali.

Tube holes should be punched $\frac{1}{8}$ in. less than tube diam. and reamed or drilled, holes being slightly countersunk on both sides. Finished holes to be from $\frac{1}{64}$ to $\frac{1}{16}$ in. larger than tube, according to size. If copper ferules are used, the ferules should be a neat fit in the holes. The tube sheet should be annealed after punching and before drilling, and the tube ends before setting. Tubes to project $\frac{1}{16}$ in. beyond sheet for each inch of diam. Tubes to be expanded only until tight. Ends which are exposed to direct flame must be flanged, beaded over and slightly re-expanded Copper ferules (No. 18 to No. 14 wire gauge) to be used in fire-tube boilers on ends exposed to direct heat. Stay bolts to be carefully threaded and holes tapped with a tap extending through both plates. Bolts to project $\frac{1}{5}$ diam. for riveting over. Thickness of nuts for screw stays >0.5 diam. of stay. Pitch of stays <10 in. If welding is necessary in braces and stays take strength of welded bar$=0.8\times$strength of solid bar. Brace rivets subject to oblique pull are allowed to bear only one-half the stress of seam rivets. Manholes to be flanged inwards on a radius$>3t$ and are to be reinforced by W.I. or steel rings, which are shrunk on. Domes when unavoidable to be flanged down to shell, and the shell to be flanged up inside the dome or else reinforced by a collar flanged at the joint, flanges being double-riveted. Drums to be put on with steel collar flanges$>\frac{3}{8}$ in. thick, double-riveted to shell and drum and single-riveted to neck or leg, or, the flanges may be formed on the legs.

Safety factors rivet seams, 4.5; flat surfaces, bumped heads, stay-bolts, braces and stays, 5. Hydrostatic test pressure should not exceed the working steam pressure by more than $\frac{1}{3}$ of itself, and this excess should not be greater than 100 lbs. per sq. in. The temperature of testing water should not be less than 125° F.

Board of Trade (B. T.) and U. S. Statute Proportions and Rules. Materials. Shells; (B.T.) T.S. from 27 to 32 tons, elongation in 10 in.$>18\%$ (if annealed, $>20\%$); 2 in. strips to stand bending until sides are parallel and not $>3t$ apart. (U.S.) When t=or<0.5 in., contraction must be= or$>50\%$, from 0.5 to 0.75 in.,$>45\%$ and above 0.75 in.,$>40\%$.

Stays (B.T.). Same T.S. as shells, elongation in 10 in.$>20\%$. Steel stays welded or worked in fire not to be used. Allowable load=9,000 lbs. per sq. in. on net section. (U.S.) Reduction of area must be$>40\%$ if test bar is>0.75 in. in diam. Allowable load=6,000 lbs. per sq. in.

Notation for the following Boiler Proportions D=boiler diam., t= thickness, t_1=thickness in 16ths, p=greatest pitch between stays, L= pitch of flanges, d=outside diam. of tubes, W=width of flame box, l_1= length of girders, p_1=pitch of bolts, D_2=distance between centers of girders, d_1=depth of girders, t_2=sum of girder thicknesses, D_3=least horizontal distance between centers of tubes, d_2=inside tube diam , W_1= width of combustion box from tube-plate to back of fire-box; all in inches. P and T are working pressure and tensile strength in lbs. per sq. in., S=surface supported in sq. in., D_1=outside flue diam. in ft., l= length of furnace (up to 10 ft.) in feet, F=safety factor,=4.5, B=percentage of strength of joint compared to solid plate.

Boiler Shells (B.T.). $P=2BTt \div DF$. (U.S.) $P=Tt \div 3D$ for single-riveting. Add 20% for double-riveting.

Flat Plates (B.T.). $P=C(t_1+1)^2 \div (S-6)$.

$C=125$ for plates not exposed to heat or flame, stays fitted with nuts and washers, the latter at least $3\times$diam. of stay and having a thickness$=\frac{2}{3}t$ of plate.

$=187.5$, same, but with diam. of washers$=\frac{2}{5}$ pitch of stays, and of thickness not less than that of the plate.

$=200$, same, but with doubling plates in place of washers, whose width$=\frac{2}{3}\times$pitch of stays, and thickness=that of plate.

=112.5, same, but stays fitted with nuts only.

$C=75$ for plates exposed to heat or flame, steam being in contact with the plates, stays fitted as where $C=125$, above.
$=67.5$, same condition, but stays fitted with nuts only.
$=100$ for plates exposed to heat or flame, water being in contact with the plates, stays screwed into plates and fitted with nuts.
$=66$, same condition, but stays with riveted heads.
(Above values for steel plates; for iron plates take 80% of same.)
(U.S.) $P=Ct_1 \div p^2$.
$C=112$ for plates $\frac{7}{16}$ in. and under, with screw stay bolts and nuts, with plain bolt fitted with single nut and socket, or with riveted head and socket.
$=120$ for plates thicker than $\frac{7}{16}$ in. for same fastenings.
$=140$ for flat surfaces, stays fitted with inside and outside nuts.
$=200$, same as for $C=140$, but with the addition of washer riveted to plate, whose thickness is at least $0.5t$ of plate and whose diam. $=0.4 \times$ pitch of stays.

N.B. Plates fitted with double angle-irons and riveted to plate with leaf at least $\frac{2}{3}t$ of plate and depth at least $\frac{1}{4} \times$ pitch are to be allowed the same pressure as that determined for plate with washer riveted on.

No brace or stay bolt in a marine boiler to have a pitch greater than 10.5 in. on fire-boxes and back connections.

Plates for Flanging (B.T.). $P=\frac{3{,}300t}{d}\left(5-\frac{L+12}{60t}\right)$. This formula is for the strength of furnaces stiffened with flanged seams where $L<120t-12$, the flanges being properly designed and formed at one heat.

Furnace Flues. Long furnaces (B.T.). $P=Ct^2 \div (l+1)D_1$, where $l>(11.5t-1)$. $C=88{,}000$ for single-strap butt-joints single-riveted, $=99{,}000$ for welded joints or butts with single straps double-riveted, and also for double-strap butt joints single-riveted.

P from above formula should not exceed the value given by the following formula for short and patent furnaces.

Short Furnaces, Plain and Patent (B.T.). $P=ct \div D_1$, where $c=$ 8,800 for plain furnaces; $=14{,}000$ for Fox (max. and min. $t=\frac{5}{8}$ and $\frac{5}{16}$ in. and plain part <6 in. long); $=13{,}500$ for Morison, same conditions as Fox; $=14{,}000$ for Purves-Brown (max. and min. $t=\frac{5}{8}$ and $\frac{7}{16}$ in., plain part <9 in. long).

Long Furnaces (U.S.). $P=89{,}600t^2 \div lD_1$ (l not to exceed 8 ft.).

Short Furnaces (U.S.). $P=ct \div D_1$, where $c=14{,}000$ for Fox ($D_1=$ mean diam.); $=14{,}000$ for Purves-Brown ($D_1=$flue diam.); $=5{,}677$ for plain flues >16 in. diam. and <40 in. diam. when not over 3-foot lengths.

Stay Girders (B.T.). $P=Cd_1^2t_2 \div (W-p_1)D_2l_1$, where $C=6{,}600$ for 1 bolt, $=9{,}900$ for 2 or 3 bolts and $=11{,}220$ for 4 bolts.

Tube Plates (B.T.). $P=20{,}000t(D_3-d_2) \div W_1D_3$. Crushing stress on tube plates caused by pressure on top of flame-box to be $<10{,}000$ lbs. per sq. in.

Air Passages through grate bars should be from 30 to 50% of grate area, the larger the better, in order to avoid stoppage of air supply by clinker, but with clinkerless coal much smaller areas may be used.

COMBUSTION.

Combustion or burning is rapid chemical combination accompanied by heat and sometimes light, during which heat is evolved equal to that required to separate the elements.

In the burning of a simple hydrocarbon (e.g., marsh gas), the combustion being complete,

$$\begin{array}{ccccccc} \text{Marsh Gas} & + & \text{Oxygen} & = & \text{Carbon Dioxide} & + & \text{Water (Steam)}; \\ CH_4 & + & 2O_2 & = & CO_2 & + & 2H_2O \end{array}$$

or, taking the atomic weights of C, H, and O as 12, 1, and 16, respectively,

$$\begin{array}{lccccc} & (12+4)+2(16\times 2) & = & [12+(16\times 2)] & + & 2(2+16), \\ \text{i.e.,} & 16\text{ lb.} + 64\text{ lb.} & = & 44\text{ lb.} & + & 36\text{ lb.} \\ \text{or} & 1\text{ lb.} + 4\text{ lb.} & \text{yields} & 2.75\text{ lb.} & + & 2.25\text{ lb.} \end{array}$$

Also, 1 lb. C burnt to CO_2 yields 14,600 B.T.U. and 1 lb. H burnt to H_2O yields 62,000 B.T.U., and, as 1 lb. $CH_4=\frac{3}{4}$ lb. C$+\frac{1}{4}$ lb. H, then

$$\begin{array}{lll} 0.75\text{ lb. C+O} & \text{yields} & 14{,}600\times 0.75=10{,}950\text{ B.T.U.} \\ 0.25\text{ '' H+O} & \text{''} & 62{,}000\times 0.25=\underline{15{,}500}\text{ ''} \\ & & \text{Total}=26{,}450\text{ ''} \end{array}$$

Experimentally, about 2,800 B.T.U. less are obtained, the loss being required to effect the work of decomposing the C and H.

Good, dry bituminous coal contains on the average, by weight Carbon, 83.5%; Hydrogen, 4.6%; Oxygen, 3.15%; Nitrogen and Sulphur (inactive elements), 8.75%.

In 100 lbs. of fuel the 3.15 lb. O is already united to ($\frac{1}{8}\times3.15$) 0.4 lb. H in the form of water, consequently this H does not assist in combustion. This leaves 83.5 lb. C and 4.2 lb. H to be dealt with.

Now, 12 lb. C unite with 32 lb. O to form CO_2 (1:2.66) and 2 lb. H unite with 16 lb. O (1:8) to form H_2O. Consequently

83.5 lb. C require $83.5\times2.66=222$ lb. O
4.2 " H " $4.2\times8 = 33.6$ " "

Or, for 100 lb. coal, total = 255.6 " "

Air = 23% O + 77% N; therefore 23 100::255.6÷100 11.1, or 11.1 lb. of air are theoretically needed for the combustion of 1 lb. of the coal. (In practice the theoretical amount must be multiplied by 1.5 for gas furnaces, by 1.5 to 2 for good grates, and by 3 or more for defective furnaces.) Also,

0.835 lb. C × 14,600 = 12,191 B.T.U.
0.042 " H × 62,000 = 2,604 "

Total B.T.U. per 1 lb. coal = 14,795

The Calorific Value of a Given Fuel may be expressed by the following modification of Dulong's formula:

B.T.U. per lb. $= 14{,}600\,C + 62{,}000\left(H-\frac{O}{8}\right) + 4{,}000\,S$, where the proportions of C, H, O, and S are determined by analysis.

Where a complete analysis of the coal is not obtainable the following formula of Otto Gmelin may be used B.T.U. per lb. $=144[100-(w+a)]-10.8\,wc$, where w and a are the percentages of water and ash, and c is a constant varying with the amount of water. When $w<3\%$, $c=4$; when w is between 3 and 4.5%, $c=6$; w bet. 4.5 and 8.5%, $c=12$; w bet. 8.5 and 12%, $c=10$; w bet. 12 and 20%, $c=8$; w bet. 20 and 28%, $c=6$; $w>28\%$, $c=4$. Also, when C and C_1 are the percentages of fixed and volatile carbon, respectively, and H the percentage of hydrogen, B.T.U. per lb. $=(14{,}600\,C+20{,}390\,C_1+62{,}000\,H)\div100$.

American Coals. Approximate Analyses and Calorific Values.

	Moisture.	Volatile Matter.	Fixed Carbon.	Ash.	Sulphur.	B.T.U. per Lb. Coal.
Anthracites:						
* E. middle field, Pa..	4.12	3.08	86.38	5.92	0.49	13,578
* N. " " "..	3.42	4.38	83.27	8.20	.73	13,434
W. " " "..	3.16	3.72	81.14	11.08	.90	12,958
Semi-anthracite:						
Loyalsock, Pa.	1.3	8.10	83.34	6.23	1.03	14,247
Semi-bituminous.						
* Clearfield, Pa.	.81	21.10	74.08	3.36	.42	14,985
* Cumberland, Md...	.95	19.13	72.70	6.40	.78	14,461
* Pocahontas, Va. ...	.85	18.60	75.75	4.80	.62	14,854
* New River, W. Va..	.76	18.65	79.26	1.11	.23	15,429
Bituminous:						
* Youghiogheny, Pa..	1.03	36.49	59.05	2.61	1.81	14,262
Connellsville, Pa.	1.26	30.10	59.61	8.23	.78	13,946
Brazil, Ind.	8.98	34.49	50.30	6.28	1.39	12,356
* Big Muddy, Ill.	7.7	31.9	53	7.4	....	12,895
Streator, Ill.	8.3	37.63	45.93	8.14	....	12,047
Rosyln, Wash. ...... (Cle-Elum.)	6.34	37.86	48.30	7.59	.49	12,429
Cokes:						
Connellsville, Pa.	(B.T.U.	pr. lb. =	88.96	9.74	.81	12,988
Chattanooga, Tenn...	%C×	14,600)	80.51	16.34	1.595	11,754
Birmingham, Ala. ...	....	.....	87.29	10.54	1.195	12,744
Pocahontas, Va.	....	.....	92.53	5.74	.597	13,509

Coals marked * are generally selected for boiler tests on account of availability, excellence of quality, and adaptability to various kinds of furnaces, grates, boilers, and methods of firing.

The number of B.T.U. per lb. of coal is calculated by means of Goutal's formula: B.T.U. per lb. of coal $=14{,}760C+aV$, where C = percentage of fixed carbon in the coal, V = percentage of volatile matter in the coal, and a = a variable depending on the ratio V_1 of the volatile matter to the amount of combustible in the coal.

Values of a:

$V_1=V\div(V+C)=$	0.05	0.1	0.15	0.20	0.25
a .... =	26,100	23,400	21,060	19,620	18,540
V_1 .... =	0.30	0.35	0 365	0.385	0.40
a .... =	17,640	16,920	16,480	15,000	14,400

This formula is fairly accurate where the percentage of fixed carbon is above 60; whenever exact results are required a calorimetric determination of the heating value of the particular fuel should be made.

Wood. 1 cord = 128 cu. ft., about 75 ft. of which are solid wood. 2.25 lbs. of dry wood are about equal to 1 lb. of soft coal in heating effect. Average wood (perfectly dry) has a calorific value of about 8,200 B.T.U. per lb.; if ordinary, air-dried (25% moisture), about 5,800 B.T.U. per lb.

Petroleum. Average composition $=0.847$ C $+0.131$ H $+0.022$ O. Sp. gr. = 0.87. B.T.U. per lb. = 20,318 (Beaumont, Tex., crude oil, 18,500 B.T.U.).

Distillates from Petroleum ($C_{10}H_{24}$ to $C_{32}H_{64}$) vary from 71.42 to 7 .77% C, and from 28.58 to 26.23% H. Sp. gr. = 0.628 to 0.792. Boiling-point varies from 86° to 495° F. B.T.U. per lb., from 27,000 to 28,000.

Gas Fuels (B.T.U. per 1,000 cu. ft.): Natural gas, 1,100,000; coal-gas, 640,000 to 675,000; water-gas, 290,000 to 327,000; gasoline-gas, 517,000; producer-gas,—anthracite, 137,000; bituminous, 156,000.

Miscellaneous Fuels (B.T.U. per lb.): Spent tanbark, 4,280 (30% water) to 6,100 (dry); straw, 5,400 to 6,500; bagasse (sugar-cane refuse), 3,750, when fibre = 45%; corn, 7,800 (ordinary condition) to 8,500 (dry).

Draft. Chimneys.

	Kent.	Gale-Meier.	Ing. Taschenbuch.
Area, A .... =	$\dfrac{0.06F}{\sqrt{h}}$	$0.07F^{\frac{3}{4}}$	Diam., $d=0.242F^{0.4}$
Height, h .... =	$\left(\dfrac{0.06F}{A}\right)^2$	$\dfrac{120}{t}\left(\dfrac{F}{G}\right)^2$	$0.216\left(\dfrac{F}{G}\right)^2+6d.$

Where F = total coal burnt per hour in lbs., t = temp. of discharge gases in F.°, G = sq. ft. of grate area, d = internal diam. in feet (A in sq. ft., h in ft.). The larger results obtained from the Taschenbuch formulas are probably due to the inferior evaporative power of German coals.

Intensity of Draft (f). f in inches of water $=h\left(\dfrac{7.64}{T_2}-\dfrac{7.95}{T_1}\right)$, where T_2 and T_1 are respectively the absolute temperatures of the external air and the chimney gases. f at the base of ordinary chimneys ranges from 0.5 to 0.75 in. In locomotives the vacuum induced by the steam-blast varies from 3 to 8 inches of water in the smoke-box and is about $\frac{1}{3}$ as much in the fire-box. The best value of $T_1=2T_2$, or about 585° F.

Temperature of Chimney Gases. To determine same approximately, suspend strips of the following metals in the chimney and note those which melt.

Metal...........	Sn	Bi	Pb	Zn	Sb
Melting-point, F.°..	456	518	630	793	810

Velocity of Chimney Gases.

$$v \text{ in ft. per sec.} = \frac{8\sqrt{h}(\text{chimney temp.—air temp.})}{3.3\times\text{chimney temp.}}$$

(Temp. in F.°).

Draft Pressures required for Combustion of Fuels (in inches of water). Wood, 0.2 to 0.25; sawdust, 0.35 to 0.5; do., with small coal,

0.6 to 0.75; steam coal, 0.4 to 0.75; slack, 0.6 to 0.9; do., very small, 0.75 to 1.25; semi-anthracite, 0.9 to 1.25; anthracite, 1.25 to 1.5; do., slack, 1.3 to 1.8.

Rate of Combustion (lbs. of fuel per hour per sq. ft. of grate area). Anthracite, 5 to 15; bituminous, 4 to 26. Ordinary combustion may be increased 50% by means of artificial draft. In locomotives the rate of combustion ranges from 45 to 85 and even 120 lbs. Low-grade or refuse fuels may be utilized with artificial draft, the high rate of combustion compensating for the low evaporative power of the fuel.

Mechanical Stoking. In the Jones underfeed stoker coal is fed into a hopper and pushed forward from the bottom thereof by a steam-actuated plunger into the retort or fire-box from beneath, air being introduced at the top of retort. As the fresh coal approaches the fire from beneath its gases are liberated by the heat and pass upwards through the fire and are consumed,—aiding in the production of heat,—and the coal reaches the fire practically coked, the production of smoke being thus avoided. The manufacturers (Underfeed Stoker Co., Ltd., Toronto) claim that its use will effect a saving of from 18 to 25% of the fuel as compared with hand-firing.

BOILER ACCESSORY APPARATUS.

Feed-Water Heating obviates in large measure the strains that would otherwise be induced by introducing water into the boiler at ordinary temperatures, and also affords considerable economy.

Saving in per cent by heating feed-water with exhaust steam $=\frac{h_2-h_1}{H-h_1}$, where H=total heat of 1 lb. steam at boiler pressure, h_1=total heat of 1 lb. water before entering heater, and h_2=same after leaving heater.

For average conditions there is an approximate saving of 1% for each increase of 11° in the temp. of feed-water, which may be heated as high as 210° F.

Green's Economizer is a feed-water heater composed of tubes so situated in the flues between boiler and chimney as to intercept some of the heat of the waste gases. As the temperature of steam from 100 to 200 lbs. pressure ranges from 338° to 388° F., all heat in chimney gases above these temperatures is wasted unless a portion of it can be absorbed in some such manner. Average chimney temps. reach 600° F.

Economizers effect a gain in evaporative power of from 6 to 30%, fair results being set at 10 to 12%, with a cooling of flue gases of from 150° to 250° F.

Condensers. In condensing the exhaust steam from an engine a partial vacuum is formed and the gain in power may be based on the increase of the mean effective pressure by about 12 lbs. per sq. in.

Jet Condensers, in which the exhaust is met by a spray of cooling water, should have a capacity of from $\frac{1}{2}$ to $\frac{3}{4}$ that of the low-pressure cylinder. Quantity of water required=25 to 30×wt. of steam to be condensed. Temp. of hot-well=110° to 120° F.

Surface Condensers should have vertical brass tubes for maximum efficiency and the water should flow downwards through them. Tubes should be as long as practicable and of small diam. (0.5 to 1 in.). Cooling surface of tubes=1 to 3 sq. ft. per I.H.P., according to climate. 12.5 lbs. steam condensed per sq. ft. per hour is good practice. Q of circulating water=30×wt. of steam condensed.

Q for jet condenser in lbs. $=\frac{H-t}{t-t_1}$, Q for surface condenser $=\frac{H-t}{t_2-t_1}$, where H=1,114° F.=total heat of 1 lb. exhaust steam, t=temp. of hot-well in F.°, t_1=entering temp. of cooling water, and t_2=temp. of water when leaving the condenser. Area of injection orifice=lbs. water per min.÷650 to 750, or, =area of piston÷250.

Evaporative Condensers. In these the exhaust is led through a large number of pipes cooled externally by trickling streams of water. This water evaporates, thus condensing the exhaust steam in the pipes, which is then pumped back into the boiler. Used where economy in water consumption is imperative. In well-designed condensers of this

class 1 lb. of water will condense 1 lb. of steam, as against the 20 to 30 lbs. of water required in jet and surface condensers.

Air-Pumps in all condensers abstract the water of condensation and the air it originally contained when entering the boiler. In jet condensers they also pump out the condensing water and its content of air. The size of an air-pump is calculated from these conditions, allowances being made for efficiency. Volume of Air-Pump in cu. ft. $= \frac{c}{n}(q+Q) = \frac{\text{I.H.P.}}{\text{r.p.m.}} \times c_1$, where n = number of useful strokes per min., q = cu. ft. of water condensed per min., Q = cu. ft. of cooling water per min., $c = 2.8$ for single-acting and 3.5 for double-acting pumps. (For jet condensers only.) $c_1 = 0.41$ for single-acting pump and jet condenser, $= 0.17$ for single-acting pump and surface condenser, and $= 0.27$ for double-acting horizontal pump and jet condenser. Vol. of single-acting air-pump = Vol. of low-pres. cyl. ÷ 23.

Circulating Pumps. Capacity $= Q \div n$. Diam. of cylinder in inches $= 13.55\sqrt{Q \div (n \times \text{length of stroke in feet})}$. (For Q and n see Air-Pumps.) The area through valve-seats and past the valves should be large enough to permit the full quantity of condensing water to flow at a velocity $\leqq 400$ ft. per min.

Fusible Plugs are screwed into those portions of boilers where the heating surface first becomes exposed from lack of water. They have a core of fusible metal at least 0.5 in. diam. tapered to withstand internal pressure. The U. S. Gov't specifies Banca tin which melts at 445° F. (2 Tin + 1 Bismuth melts at 334° F., 3 Tin + 1 Bismuth at 392° F.).

Safety-Valves, Area (U.S.). Lever valves: area = 0.5 sq. in. per sq. ft. of grate area. Spring-loaded valves; $\frac{1}{3}$ sq. in. per sq. ft. of grate area. Spring-loaded valves for water-tube, coil, and sectional boilers carrying over 175 lbs. pressure must have an area $> \frac{1}{6}$ sq. in. per sq. ft. grate area. Seats to be inclined 45° to axis. Spring-loaded valves to be supplied with a lever which shall raise valve from seat to a height equal to at least $\frac{1}{8}$ diam. of opening.

(B.T.) Area in sq. in. = (37.5 × grate area in sq. ft.) ÷ (gauge pressure + 15). Philadelphia Rule: Area in sq. in. = (22.5 × grate area in sq. ft.) ÷ (gauge pressure + 8.62). Ingenieurs Taschenbuch $a = 0.0644\sqrt{\frac{V}{p}}$, where a = area of valve in sq. in. per sq. ft. of heating surface, p = max gauge pressure, V = cu. ft. of steam per lb. at pressure p.

Injectors (Live-Steam). Water injected in gals. per hour $= 1{,}280D^2\sqrt{P}$, where D = diam. of throat in ins., and P = steam pressure in lbs. per sq. in.

$$\text{Area of narrowest part of nozzle in sq.in.} = \frac{\text{cu.ft. of feed-water per hour (gross).}}{800\sqrt{\text{Pressure in atmospheres}}}$$

One lb. steam will inject about 14 lbs. water. An exhaust-steam injector will feed against pressures < 80 lbs., the feed being at about 65° F. An auxiliary live-steam jet can be attached to feed against 110 lbs. pressure, and, by compounding another live-steam injector with it, a boiler may be fed up to about 200 lbs. pressure, the feed reaching boiler in this case at about 250° F.

Injector vs. Pump. Saving of fuel over amount required when a direct-acting pump feeds at 60° F. (without heater, boiler evaporating 10 lbs. water at 212° F. per lb. of fuel).

Injector feeding at 150°, no heater,	saving,	1.5%
" " through heater (from 150° to 200°),	"	6.2%
Direct-acting pump through heater (from 60° to 200°),	"	12.1%
Geared " " " (" 60 " 200°),	"	13.2%

Steam-Pipes (B.T.). d = inside diam., t = thickness, both in inches; p = pressure in lb. per sq. in.

Copper Pipes. brazed, $t = \frac{pd}{6{,}000} + \frac{1}{16}$ in.; solid-drawn, $t = \frac{pd}{6{,}000} + \frac{1}{32}$ in.

Lap-welded Iron Pipes, $t = \frac{pd}{6{,}000}$; Cast-iron Pipes, $t = \frac{pd}{3{,}500} + \frac{1}{4}$ in.

Provision should be made for expansion in long lines, which amounts to about 1 in. in 50 ft. for the range of temperatures usually employed.

INCRUSTATION AND CORROSION.

Incrustation or scale is the hard deposit in boilers resulting from the precipitation of impurities from water boiling at high temperatures. Scale of $\frac{1}{16}$ in. thickness will reduce boiler efficiency $\frac{1}{8}$, and the reduction of efficiency increases as the square of the thickness of scale. A larger amount than 100 parts in 100,000 of total solid residue will generally cause troublesome scale, and waters containing over 5 parts in 100,000 of nitric, sulphuric, or muriatic acids are liable to cause serious corrosion.

Prevention and Cure of Boiler Troubles due to Water.

Trouble.	Troublesome Substance.	Remedy or Palliative.
Incrustation. .	Sediment, mud, clay, etc.	Filtration, blowing-off.
	Readily soluble salts.	Blowing-off.
	Bicarbonates of magnesia, lime, and iron.	Heating feed and precipitating by addition of caustic soda, lime, magnesia, etc.
	Sulphate of lime.	Addition of carbonate of soda or barium chloride.
Priming......	Carb. of soda in large amounts.	Addition of barium chloride.
	Organic matter (sewage).	Precipitate with alum or ferric chloride and then filter.
Corrosion.....	Organic matter.	Ditto.
	Acid in mine waters.	Add alkali.
	Dissolved carbonic acid and oxygen.	Heating feed, addition of caustic soda, slacked lime, etc.
	Grease.	Slacked lime and filtering. Carb. of soda. (Substitute mineral oils.)

Many scale-making minerals may be removed by using a feed-water heater and employing temperatures at which the minerals are insoluble and consequently precipitate, when they may be blown off before passing to boiler. Phosphate of lime, oxide of iron and silica are insoluble at 212° carbonate of lime, at 302°, and sulphate of lime at 392° F

Kerosene has been successfully used in softening and preventing scale and should be introduced into the feed-water in quantities not exceeding 0.01 qt. per H.P. per day of 10 hours.

Tannate of Soda Compound.—Dissolve 50 lb. sal soda and 35 lb. japonica in 50 gal. water, boil and allow to settle. Use $\frac{1}{40}$ qt. per H.P. per 10 hours, introducing same gradually with the feed-water.

Grooving is the cracking of plate surface due to abrupt bending under alternate heating and cooling. It is generally found near rigid stays and its ill effects are augmented by corrosion. It may be avoided by providing for sufficient elasticity along with strength and by rounding the stay edges at the plate.

INTERNAL-COMBUSTION ENGINES.

Internal-combustion engines are divided into two classes. In the first an explosive charge of gas and air (or a vapor of alcohol, gasoline, or kerosene, mixed with air) is drawn into the cylinder, compressed, ignited, expanded, and then exhausted. The ignition produces a practically instantaneous explosion.

In the second class (e.g., Diesel motors) a charge of air is drawn in and is raised by compression to a temperature high enough to ignite the oil, gasoline or other fuel which is sprayed into the cylinder during a certain portion of the power stroke. The combustion in this case is gradual and extends over the period of the stroke during which the fuel is injected

In simple engines there are four strokes in the cycle of operation 1st stroke, drawing in of explosive charge; 2d (return) stroke, compression of the charge; 3d stroke, ignition and expansion (power stroke); 4th (return) stroke, exhaust of the burnt gases. The 1st, 2d, and 4th strokes consume from 5 to 10% of the power developed on the 3d stroke. (For indicator card, see Fig. 12, *T*.)

In two-cycle engines the charge is compressed in a separate cylinder, ignition and expansion taking place on the 1st or outward stroke of engine and exhaust on the return stroke,—there being one impulse for each revolution of fly-wheel. Large engines are also constructed so as to give an impulse on each stroke.

Fuels. The thermal efficiency of an internal-combustion engine is increased by high compression, the only limit being that the temperature at the end of the compression must not approach near to that of ignition. The temperature of ignition varies inversely as the number of B.T.U. contained in the charge, and rich gases, therefore, should not be highly compressed save in well diluted charges. The limits of compression may be extended by cooling the gases undergoing compression, as in the Banki motor, where water is sprayed into the cylinder to absorb the heat given out during compression, and also as in the Diesel engine, where the air is compressed to its final pressure before the fuel is injected.

Rich Gases (containing over 350 B.T.U. per cu. ft.). Coal, coke-oven, and natural gases.

	Rich Mixture.	Lean Mixture.
Ratio of gas to air.	1:6 to 1:7	1:10 to 1:15
Temperature of ignition. . . .	1,000 to 1,100° F. abs.	1,200 to 1,380° F. abs.
Compression, lbs. per sq. in..	55 " 70	75 " 115
M.E.P. " " " "	70 " 85	65 " 78
Explosion pressure per sq. in.	210 " 285	285 " 355

Lean Gases (containing less than 350 B.T.U. per cu. ft.). Dowson, producer, and blast-furnace gases.

Ratio of gas to air.	1:1 to 1:2
Temperature of ignition.	1,300 " 1,475° F. abs.
Compression.	115 " 215 lbs. per sq. in.
Mean effective pressure.	65 " 78 " " " "
Explosion pressure.	255 " 355 " " " "

The gas and air should be thoroughly mixed before ignition, which, for rich mixtures, is either by a hot tube, a valve-governed flame, or by an electric spark. For lean mixtures the electric spark is used.

Liquid Fuels.

	Gasoline, Benzine.		Kerosene, Naphtha, Alcohol.
Ignition temperature, ° F. abs.	930 to 1,020		985 to 1,075
Compression, lbs. per sq. in.	40 " 70		55 " 115
" (Banki motor).	170 " 210	(Diesel)	450 " 500
Explosion pressure, lbs. per sq. in. . . .	170 " 285		140 " 285
" " (Banki).	565		
M.E.P., lbs. per sq. in.	57 " 78		50 " 70

Liquid fuels are vaporized before mixing. Light oils (gasoline, etc.) are vaporized by the heat of the air drawn through or over them, or they may be atomized. Heavier oils require heating in order to vaporize. Gasoline-gas is usually ignited by an electric spark,—heavier oils by the hot tube.

Average Values for Compression (Lucke). Kerosene and city gas, 80 lbs.; gasoline, 85 lbs.; natural gas, 115 lbs.; producer gas, 135 lbs.; blast-furnace gas, 155 lbs. (All pressures are absolute.)

Fuel Consumption (C_h) **per B.H.P. Hour,** and actual thermal efficiencies (η_w).

		5 H.P.		25 H.P.		100 H.P.	
		C_h	η_w	C_h	η_w	C_h	η_w
Coal gas,	cu. ft. . .	19	0.20	15.5	0.24	13.8	0.27
Producer gas,	" " . .	105 to 115	0.17	85 to 92	0.21	75 to 80	0.24
Blast-furnace gas,	" " . .			115	0.20	100	0.24
Coke-oven gas,	" " . .			30	0.19	24.7	0.23
Gasoline,	lbs. . .	0.66	0.19	0.55	0.23		
Kerosene,	" . .	1.2	0.11	1.02	0.13		
Alcohol, 90%	" . .	1.1	0.22	0.92	0.26		
Petroleum, crude, (Diesel motors)	" . .	0.55	0.25	0.51	0.27	0.44	0.315

Properties of Fuels.

	B.T.U. per Cu. Ft. (H.)	Lbs. per Cu. Ft. (Atmos.	Cu. Ft. per Lb. Pressure.)	Cu. Ft. Air Required for Combustion of 1 Cu. Ft. Gas. Theoret.	Actual.
Coal-gas, average. . .	650	.035	28.5	5.6 to 6.5	9 to 10
" N. Y. City.	710 to 720				
Producer-gas:					
Anthracite.	140	.062	16	.85	1.1
		to	to	to	to
Coke.	130	.075	13.5	1	1.4
Water-gas (coke). . .	275	.044	22.7	2.4	3 to 4
Blast-furnace gas. . .	106	.08	12.4	.75	1 to 1.2
Coke-oven gas.	450	.042	24	5.3	7
Natural gas.	1,000 to 1,100	.0458	21.83	9	12.5
do. Pittsburgh . . .	495 to 585				
* Acetylene.	1,550	. . . . . .	. . . . .	12.5	18 to 20
	B.T.U. per Lb.			Cu. Ft. Air per Lb. Fuel.	
Petroleum.	18,500	50	.02	185	250 to 350
(Kerosene)	22,000	. . . .			
Benzine, gasoline. . .	18,000–20,000	43.8	.0229		240 to 320
Alcohol, grain (90%)	10,900	51.9	.019	96	125 to 190
" wood.	8,300				

* One pound of calcium carbide liberates 5.75 cu. ft. of acetylene gas.

Cooling Water (when entering cylinder jacket at about 60° F. and leaving at about 150° F.) should be supplied at the rate of 40 to 45 lbs. per hour per I.H.P. (or 5 to 5.5 gal.). Supply tanks should have a capacity of 20 to 30 gal. per I.H.P.

Efficiencies. Actual thermal efficiency, $\eta w = 2{,}545 \div HCh$. Mechanical efficiency, $\eta m = \text{B.H.P.} \div \text{I.H.P.}$ Indicated thermal efficiency, $\eta i = \eta w \div \eta m$. Theoretical thermal efficiency, $\eta_t = (1.25 \text{ to } 2)\eta i$.

Average Values of ηm (Lucke).

I.H.P. of Engine.	Four-cycle.	Two-cycle.
500 and larger. . . .	.81 to .86	.63 to .70
25 to 500.	.79 " .81	.64 " .66
4 " 25.	.74 " .80	.63 " .70

Brake Horse-Power $= aspm\eta mE \div (12 \times 33{,}000) = (\pi d^2 sE \times 65 \times 0.85) \div (4 \times 12 \times 33{,}000) = 0.0001096 d^2 sE$, where a = area of cylinder in sq.in. $= 0.7854 d^2$, s = stroke in inches, pm = mean effective pressure (average = 65 lbs. per sq. in.), $\eta m = .85$, E = number of explosions per min. = r.p.m. ÷ 2, for a four-cycle engine.

Piston Speeds. Average practice in ft. per min. $= 600 + 0.2 \times \text{H.P.}$

Valve Setting. The exhaust should close when engine is on center; the inlet should open about 5° after center is passed and continue about 10° beyond center after compression has begun.

Ratio of Clearance to Stroke $\left(\frac{c}{s}\right)$, where c = volume of clearance space in cu. in. ÷ area of cyl. in sq. in.

	$c \div s$.	Compression.
Natural gas.	0.3	100 lb. per sq. in.
Rich gas, rich mixture.	0.47 to 0.77	65 to 40 " " " "
" " , lean "	0.26 " 0.38	115 " 80 " " " "
Lean gas.	0.18 " 0.26	170 " 115 " " " "
Benzine.	0.54 " 1.44	56 " 28 " " " "
" (Banki).	0.146 " 0.177	210 " 170 " " " "
Petroleum, Alcohol.	0.42 " 0.77	70 " 42 " " " "
" (Diesel)	0.072 " 0.077	500 " 450 " " " "

Expansion and Compression Laws. $PV^n = P_1V_1^n$. For expansion n ranges from 1.25 to 1.4, and for compression, from 1.2 to 1.5. For expansion, n is generally taken at 1.35, and at 1.3 for compression. If n is taken at 1.33, the following formulas may be used:

Pressures and Temperatures (Absolute). Let P = suction pressure in lbs. per sq. in., P_c = compression pressure, Pe = explosion pressure, P_r = exhaust pressure, T = initial temperature of charge in degs. F. absolute, T_c = temp. at end of compression, Te = explosion temperature, T_r = exhaust temperature, s = stroke in in., and c = clearance expressed as inches of stroke. Then, $P_c = P\sqrt[3]{[(s+c)\div c]^4}$. T for scavenging engines $= 100\left(1+\frac{c}{s}\right) + 461$; for non-scavenging engines, $T = 120[1+(c\div s)]+461$.

$T_c = T\sqrt[4]{P_c \div P} = T\sqrt[3]{[(s+c)\div c]}$. $Te = T_c + R$ if scavenging; if not, $Te = T_c + R \div [1+(c\div s)]$, where R is the rise of temperature due to explosion and is obtained from a table which follows. $Pe = P_cTe \div T_c$. $P_r = Pe \div \sqrt[3]{\left(\frac{s_1+c}{c}\right)^4}$, where s_1 = inches of stroke completed at point of release. $T_r = Te \div \sqrt[4]{Pe \div P_r} = Te \div \sqrt[3]{[(s_1+c)\div c]}$.

Ratio of Air to Gas (volumetric), $a = (C \div 50) : 1$ for best economy, $a = (C \div 60) : 1$ for maximum possible load. C = calorific value of gas in B.T.U. per cu. ft.

Calorific Value of Explosive Mixture, $C_1 = C \div (a+1)$.

Properties of the Constituent Elements of Gases.

(32° F., atmospheric pressure.)

	Specific Heat. k_v.	Specific Heat. k_p.	Lbs. per cu. ft.	Lbs. Oxygen per lb. Gas for Combustion.	Cu. ft. Air required by 1 cu. ft. of Gas for Combustion.	B.T.U. per lb. of Constituent Gas. High.	B.T.U. per lb. of Constituent Gas. Low.
Hydrogen, H.	2.414	3.405	.00559	8	2.43	61,560	51,840
Marsh-gas, CH_4.	.470	.593	.0445	4	9.66	23,832	21,438
Ethylene, C_2H_4.	.332	.404	.0778	3.434	14.5	21,384	20,016
Carbon-monoxide, CO.	.176	.248	.0777	.571	2.41	4,392	4,392
Carbon-dioxide, CO_2. .	.154	.217	.1221			B.T.U. per cu. ft. High.	B.T.U. per cu. ft. Low.
Nitrogen, N.	.173	.244	.0778				
Oxygen, O.	.156	.218	.0888				
Air.	.169	.2377	.08011				
Gas-engine exhaust (coal gas).	.189	.258			H	344.12	289.79
					CH_4	1060.52	954
					C_2H_4	1663.68	1557.24
					CO	341.26	341.26

(Weights in above table have been calculated from the latest values given to atomic weights. The B.T.U. values have been taken from Des Ingenieurs Taschenbuch. The values for specific heat are taken from a table by W. W. Pullen, in Fowler's Pocket-Book.)

Calculation of the Calorific Value of a Gas (1 cu. ft. at 32° F.). The table on page 99 gives the calculations for a high-grade coal-gas.

The difference between the high and low values of the B.T.U. in the tables is due to the heat of condensation of that amount of steam which results from burning the hydrogen in one cubic foot of gas. The low value should be used in calculations, this being the only heat liberated in the cylinder.

	Volume in cu. ft.	Weight in lbs.	Specific Heat.		B.T.U. (Low).	Air, cu. ft. for complete Combustion.
			k.	k_p.		
H.	.3978	.00222	.1553	.2191	115.28	.967
CH_4. . . .	.4516	.02010	.2738	.3455	430.83	4.362
C_2H_4. . . .	.0638	.00496	.0477	.0580	99.35	.925
CO.	.0704	.00547	.0278	.0392	24.02	.170
CO_2. . . .	.0108	.00132	.0059	.0083		
N.	.0050	.00039	.0020	.0003	669.48	6.424
	1.0000	.03451	.5127	.6732		

$$k_p \div kv = 1.313 = n.$$

If a 10 · 1 mixture of the above gas be used in an engine the calculations are as follows: 1 cu. ft. of mixture (10 vols. air + 1 vol. gas) weighs $[(.08011 \times 10) + .03451] \div 11 = .07596$ lb. Specific heat, $kv = .1832$; $k_p = .2553$; $k_p \div kv = n = 1.394$. Heat required to raise one cubic foot 1 degree F. = .013916 B.T.U. $= h$. Heat evolved by combustion of 1 cu. ft. of mixture = 60.862 B.T.U. $= H$. $H \div h = 4{,}374°$ F. abs.

The efficiency of combustion of coal-gas has been experimentally determined to be as follows:

Ratio of mixture.	6:1	8 1	10:1	12.1
Efficiency, x.	.465	.543	.575	.580

The rise of temperature due to explosion at constant volume, $R = Hx \div h$, in this case $= 4{,}374 \times .575 = 2{,}515°$ F.

If this mixture be compressed from 15 lbs. absolute to 80 lbs. absolute, in a common or non-scavenging engine, $(s+c) \div c = 3.51$, $s = 2.51c$, $s \div c = 2.51$, and $c \div s = .4$. Substituting these values in the preceding formulas, $T = 629°$ F., $T_c = 956°$ F., $T_e = 2{,}753°$ F., $T_r = 1{,}860°$ F. $P = 15$ lb., $P_c = 80$ lb., $P_e = 231$ lb., $P_r = 47.86$ lb. (s_1 taken $= 0.9s$).

For a scavenging engine, $T = 601°$ F., $T_c = 914°$ F., $T_e = 3{,}429°$ F., $T_r = 2{,}315°$ F. $P_e = 300$ lb., $P_r = 62.3$ lb. (All pressures and temperatures are absolute.)

The Diesel Engine. Clearance $= 0.0625$ to $0.07 \times$ vol. of cyl. Compression: $PV^{1.3} = C$; expansion: $PV^{1.2} = C$. Temperature at the end of compression to 500 lbs. pressure $= 720°$ F.; temperature at the end of combustion $= 1{,}922°$ F. A test by Mr. Ade Clark in March, '03, showed a consumption of 0.333 lb. of Texas fuel oil (19,300 B.T.U. per lb.) per I.H.P., or 0.408 lb. per B.H.P. and an efficiency of 32.3%.

Various Engine Performances. Koerting engine, 900 H.P., 28% efficiency on B.H.P. (33.5% eff. I.H.P.). A Diesel engine of 160 H.P. tested by W. H. Booth used 0.45 lb. of heavy fuel oil per B.H.P. A Crossley engine using producer-gas required from 0.65 to 0.85 lb. anthracite per B.H.P. A Hornsby-Akroyd oil engine showed a consumption of 0.785 lb. of crude Texas oil per B.H.P.

Design and Proportions of Parts. The following matter is condensed from an article by S. A. Moss, Ph. D., in Am. Mach., 4-14-04. The results have been derived from 76 single-acting engines (5 to 100 H.P.) made by 20 builders and will serve as an index of average practice Maximum explosion pressures varied from 250 to 350 lbs. per sq. in., and 300 lbs. has been taken as an average. Compression varied from 50 to 100 lbs. (50 for gasoline, 100 for natural gas) and 70 lbs. has been taken as an average. Maximum H.P. was found to be about $1.125 \times$ rated H.P. Mechanical efficiency about 80%. Values to the right, in brackets, are taken from Roberts' Gas-Engine Handbook.

Diam. of cylinder in ins. . . . . $= d$.

Thickness of cylinder wall, t. . . . . $= \frac{d}{16} + 0.25$ in. [$t = 0.09d$].

" " jacket " . . . . $= 0.6t$ [$t = 0.045d$].

" " water jacket . . . . $= 1.25t$ [$t = 0.1d$]

No. of cylinder-head studs. . . . . $=0.66d+2$.
External diam. of studs. . . . . $=d\div 12$ (average).
Length of stroke l. $=1.5d$ "
" " connecting-rod, c. . . . . $=2.5l$ "
Weight of piston, w in lbs. . . . . $=1.3a$ (a=area of cyl. in sq. in.).
" " connecting-rod w_1. . . . . $=0.8a$.
" " reciprocating parts $(w+0.5w_1)$. $=w_2a$; w_2 average $=1.7$.
Length of piston trunk. . . . . $=1.5d$ (average).
Bearing pressure on piston due to weight $=0.89$ lb. per sq. in.
Thickness of rear wall of piston. . . . . $=d\div 10$.
Wrist-pin, diam . . . . $=0.22d$; length $=1.75\times$diam.
Diam. at mid-section of connecting-rod . $=0.23d$.
Crank-pin: length $=0.39d$; diam. $=0.41d$.
Crank-throws: thickness $=0.26d$; breadth $=0.55d$.
Diam. of crank-shaft, $s=0.375d$.
Main bearing, length $=0.85d$ (bearing pressure averages 125 lbs. per sq. in.).
Fly-wheel: outside diam. . . . . $=12{,}300\div N$ (N=r.p.m.).
" weight in lbs. . . . . $=33{,}000\times$H.P.$\div N$.
Revs. per min. N. . . . . $=800\div\sqrt{l}$ [$N=380\div$(B.H.P.)$^{0.21}$ for 4-cycle, increase $\frac{1}{6}$ for 2-cycle.]

Piston speed, ft. per min. . . . $=133\sqrt{l}$.
Exhaust pipe diam. . . . . $=0.28d$.
" valve " . . . . $=0.3d$ [$0.35d$].
Inlet " " . . . . $=0.27d$ [$0.316d$].
Gas pipe " . . . . $=0.11d$.
" valve " . . . . $=0.15d$.
Air pipe " . . . . $=0.25d$
Max. B.H.P. $=d^2lN\div 14{,}400$. [For gasoline, divide by 18,000 (4-cycle) or by 13,500 (2-cycle).]

M.E.P. $=50$ to 85 lbs. per sq. in.; average, 70 lbs.

Speed of exhaust gases $=5{,}200$ ft. per min. (average).
" " inlet charge . $=6{,}400$ " " " "
" " gas . . . . $=3{,}700$ " " " "
" " air . . . . $=6{,}900$ " " " "

Dr. Lucke (in "Gas-Engine Design," D. Van Nostrand Co.) states that engines should be designed to withstand max. pressures of 450 lbs. per sq. in The following additional formulas are taken from his work:

Thickness of cylinder wall, $t=(.062$ to $.075)d+0.3$ in. Wrist-pin: diam. $=0.35d$, length $=0.6d$.

Piston rings: number $=3$ to 10, width $=0.25$ to 0.75 in., greatest radial depth $=0.02d+0.078$ in. (Güldner), or, $=0.033d+0.125$ in. (Kent). Valve diam., $v=(0.3$ to $0.45)d$; valve-stem diam. $=(0.22$ to $0.3)v$; valve lift $=(0.05$ to $0.1)v$ for flat valves,—50% greater for 45° conical valves; valve-seats, width $=(0.05$ to $0.1)v$; valve-faces $=(1.1$ to $1.5)\times$width of seat, for conical valves.

The following additional data are taken from E. W. Roberts' Gas-Engine Handbook l(for two-cycle) $=d$ to $1.25d$; diam. of water-pipes $=0.15d$; diam. of fly-wheel hub $=2s$; hub length $=1.75s$ to $2.25s$; mean width of oval spoke or arm $=0.8s$ to $1.2s$; mean thickness of arm $=(0.4$ to $0.5)\times$ mean width; number of spokes $=6$ (generally).

Engine Foundations. In order to absorb the vibrations of an engine it should be bolted to a foundation whose weight F is not less than $0.21E\sqrt{N}$, where E=wt. of engine in lbs. Brick foundations weigh about 112 lbs. per cu. ft. and those of concrete about 137 lbs., an average being about 125 lbs. per cu. ft. Number cu. ft. in foundation $=F\div 125$. The inclination or "batter" of the foundation walls from top to bottom should be from 3 to 4 in. per foot of height (E. W Roberts).

AIR.

Air is a mechanical mixture of oxygen and nitrogen,—21 parts oyxgen + 79 parts nitrogen, by volume (23 parts O+77 parts N, by weight).

1 cu. ft. of pure air at 32° F. and at a barometric pressure (B) of 29.92 inches of mercury (14.7 lbs. per sq. in.) weighs 0.080728 lb., and the volume of 1 lb. $=12.387$ cu. ft. At any other temperature and pressure,

weight per cu. ft., $w=\frac{1.3302B}{461+t}=\frac{2.707p}{461+t}$, where B=height of mercury in barometer in in., t=temperature in degs. F., 1.3302=weight in lbs. of 461 cu. ft. of air at 0° F. and 1 in. barometric pressure. Air expands $\frac{1}{493}$ of its volume for each increase of 1° F., and the volume varies inversely as the pressure.

Air liquefies at −220° F. (its critical temperature) under a pressure of 573 lbs. per sq. in. and boils at −312° F. Specific gravity at −312° F. =0.94. Latent heat=123 to 144 B.T.U. per lb. Liquid air occupies about $\frac{1}{800}$ of the volume of the same weight of free air at normal temperatures.

Barometric Determination of Altitudes. Pressure of the atmosphere at sea-level (32° F.)=14.7 lbs. per sq. in. Difference of levels (at 32° F.) in feet$=60{,}463.4\log\frac{B}{B_1}$ (1), where B and B_1 are the barometric readings of the two levels. If B is taken at sea-level it is equal to 29.92 in. and Height above sea-level$=60{,}463.4\log\frac{29.92}{B_1}$ (2).

For any other temperatures, t (for B) and t_1 (for B_1), formulas (1) and (2), must be multiplied by a correction factor, $c=1+0.00102(t+t_1-64)$.

Approximately, the pressure decreases 0.5 lb. per sq. in. for each thousand feet of ascent.

Flow of Air in Pipes. Q, in cu. ft. per min.$=c\sqrt{\frac{pd^5}{wL}}$, where p=difference between the entering and leaving gauge pressures in lbs. per sq. in., d=diam. of pipe in in., L=length of pipe in feet, and w=density of the entering air (lbs. per cu. ft.).

When						
	d=1 in.	2 in.	3 in.	4 in.	9 in.	12 in.
	c=45.3	52.6	56.5	58	61	62

Richards' formula is $Q=100\sqrt{\frac{apd^5}{L}}$.

When						
	d=1 in.	2 in.	3 in.	4 in.	8 in.	12 in.
	a=0.35	0.565	0.73	0.84	1.125	1.26

Flow of Air through Orifices. Theoretical velocity in feet per sec. $v=\sqrt{2g\times27{,}816\left(1-\frac{p_1}{p}\right)}=1{,}337.7\sqrt{1-\frac{p_1}{p}}$, where p is the pressure in the reservoir out of which the air flows, and p_1 the pressure of the receiving-reservoir. For the actual efflux the value of v must be multiplied by the proper one of the following coefficients.

Pressure (in atmospheres).	0.1	0.5	1	5	10	100
Orifice in thin plate.......	0.64	0.57	0.54	0.45	0.436	0.423
" , short tube........	0.82	0.71	0.67	0.53	0.51	0.487

Loss of pressure, $p=0.107v^2wL\div c^2d$, where w at ordinary temps.$=0.03(p_1\div14.7)^{0.71}$, p_1 (at entrance, absolute) and p both in lbs. per sq. in.

COMPRESSED AIR.

Free air is that at atmospheric pressure and at ordinary temperatures (14.7 lb. per sq. in., 62° F.). Absolute pressure=gauge pressure+14.7 lb. Absolute temperature=461° F.+reading of thermometer in degs. F.

Relations between Temperature, Volume, and Pressure.

$$\frac{p}{p_1}=\left(\frac{V_1}{V}\right)^{1.41}=\left(\frac{\tau}{\tau_1}\right)^{3.44};\ \frac{V}{V_1}=\left(\frac{p_1}{p}\right)^{0.71}=\left(\frac{\tau_1}{\tau}\right)^{2.44};\ \frac{\tau}{\tau_1}=\left(\frac{V_1}{V}\right)^{0.41}=\left(\frac{p}{p_1}\right)^{0.29}.$$

$PV=R\tau$; $R=53.354$; $P=ap$. In the foregoing p, V, τ, and p_1, V_1, τ_1 are the respective initial and final absolute pressures, volumes, and absolute temperatures.

Work of Compression. Ft.-lbs. of work required to compress 1 cu. ft. of free air to any desired pressure, p_1, isothermally$=144p\times\log_e\frac{p_1}{p}$.

If $p = 14.7$ lb., work in H.P. $= 0.0641 \log_e \frac{p_1}{14.7}$, when compressed in 1 min.

Ft.-lbs. of work required to compress 1 lb. of free air adiabatically at the absolute temperature τ, $= (\tau_1 - \tau) \times 778 \times 0.2375 = 184.7(\tau_1 - \tau)$ ft.-lbs. $= 184.7\tau \left[\left(\frac{p_1}{p}\right)^{0.29} - 1 \right]$, where τ_1 is the temp. corresponding to the volume to which the air is compressed. For work to compress 1 cu. ft. divide above value by the number of cu. ft. in 1 lb. at τ.

In practice the actual work = work of isothermal compression + about 60% of the difference between isothermal and adiabatic work.

The Output of a Compressor at any Altitude expressed in per cent $= 100 - 0.0028 \times$ height in feet (approx.).

Loss by Cooling varies from 70% under bad conditions to 20% with reheating and air injection.

Loss by Pipe Friction per mile = 5%.

Reheating. Gain by reheating in per cent $= 100 \left(1 - \frac{\tau}{\tau_1}\right)$, where τ and τ_1 are the absolute temperatures before and after heating.

Tests made at Cornell University show that from 28 to 38% gain in thermal economy can be made by reheating air from 90° to 320° F., the efficiency of the reheater being 50%. There is no additional gain made by heating above 450° and if 300° is much exceeded there is danger of charring the lubricant.

Pneumatic Tools (cu. ft. of free air required per min., 80 lbs. pressure). Chipping and calking tools, 11 (light) to 17 (heavy); riveting tools, 15 (½ in. rivet) to 22 (1¼ in. rivet); drills (metal), 15 (1 in.) to 35 (3 in.); wood-boring, 12 (1 in.) to 18 (2½ in.).

FANS AND BLOWERS.

Let h = pressure generated in inches of water (1 in. water = 0.577 oz. per sq. in. 1 oz per sq. in. = 1.73 in. water); v = peripheral velocity of wheel in ft. per sec.; v_1 = velocity of air entering the wheel through the suction openings in side of case (25 to 33 ft. per sec.); d = diam. of suction openings in in. (for openings on both sides of wheel, $d = 13.54\sqrt{q \div 2v_1}$; for opening one side only, $d = 13.54\sqrt{q \div v_1}$); D_1 = inner diam. of wheel = d to $1.5d$; D = outer diam. $= 2D_1$ for suction-fans ($= 3D_1$ for blowers); N = r.p.m. $= 229v \div D$; b = width of vanes at $D_1 = 0.25d$ to $0.4d$ for suction opening on one side ($= 0.5d$ to $0.8d$ for openings on both sides); b_1 = width of vanes at D, $= bD_1 \div D$; No. of vanes $= 0.375D$; q = cu. ft. of air per sec.; η = efficiency = 0.5 to 0.7 for large fans (0.3 to 0.5 for small fans); c = 1.2 to 1.4 for large fans (1.4 to 1.7 for small fans); α = angle which the extreme outer element of a vane makes with the radius at that point. Then, $v = 3.28[4 \tan \alpha + \sqrt{(4 \tan \alpha)^2 + 200h}]$. α is positive when the vanes are curved or inclined backward from the direction of rotation (negative when forward). For radial vanes $\alpha = 0$, and $v = 46.4c\sqrt{h} = 46.4\sqrt{h \div \eta}$. Area of discharge-opening in sq. in. $= 144\ q \div v_2$, where v_2 = velocity of air in pipe in ft. per sec. H.P. required $= qh \div 105.7\eta$. Outer diam. of disc fan in in. $= 3\sqrt{q}$; $\eta = 0.2$ to 0.3.

MECHANICAL REFRIGERATION.

Mechanical refrigeration is produced by expanding a heat medium from a normal temperature to one which is below the usual limits for the climate and zone where the expansion takes place. Media are chosen with regard to their willingness to surrender their heat energy to surrounding objects, and vapors are therefore best employed.

The vapor chosen is compressed and then relieved of its heat in order to diminish its volume. It is then expanded so as to do mechanical work and its temperature is lowered. The absorption of heat at this stage by the vapor in resuming its original condition constitutes the refrigerating effect.

Ammonia (NH_3), Sulphur dioxide (SO_2), Pictet fluid (SO_2+3% of carbonic acid, CO_2) and air are most employed, ammonia and air being of principal importance. Air is used on shipboard where pungent vapors would be objectionable.

Air. $\left(\frac{V_1}{V}\right)^{0.41} = \left(\frac{p}{p_1}\right)^{0.29} = \frac{\tau}{\tau_1}$. Air is cheap and harmless, but its use is limited on account of its bulk and the size of the machinery employed. Efficiency, measured in ice-melting effect (latent heat of fusion of ice = 142.2 B.T.U.) is between 3 and 4 lbs. of ice-melting capacity per lb. of fuel, assuming 3 lbs. of fuel per H.P.

Saturated Ammonia is inexpensive, remains liquid under atmospheric pressure only below −30° F., and at 70° F. under 115 lbs. gauge pressure.

Properties of Saturated Ammonia.

Temp. Degs. F.	Abs. Pressure, Lbs. per Sq. In.	Heat of Vaporization, B.T.U.	Vol. of Vapor. Cu. Ft. per Lb.	Vol. of Liquid. Cu. Ft. per Lb.	Wt. in Lbs. of 1 Cu. Ft. of Vapor.
−40	10.69	579.67	24.38	0.0234	0.0411
−30	14.13	573.69	18.67	.0237	.0535
−20	18.45	567.67	14.48	.0240	.0690
−10	23.77	561.61	11.36	.0243	.0880
0	30.37	555.5	9.14	.0246	.1094
+10	38.55	549.35	7.20	.0249	.1381
20	47.95	543.15	5.82	.0252	.1721
30	59.41	536.92	4.73	.0254	.2111
40	73	530.63	3.88	.0257	.2577
50	88.96	524.30	3.21	.0261	.3115
60	107.60	517.93	2.67	.0265	.3745
70	129.21	511.52	2.24	.0268	.4664
80	154.11	504.66	1.89	.0272	.5291
90	182.8	498.11	1.61	.0274	.6211
100	215.14	491.5	1.36	.0277	.7353

Ammonia Compression System. The ammonia vapor is compressed to about 150 lb. pressure and a temp. of 70° F., and is then allowed to flow into a cooler or surface-condenser, where the heat due to the work of compression is withdrawn by the circulating water and the vapor is condensed to a liquid. It is then allowed to pass through an expansion cock and to expand in the piping, thereby withdrawing heat from the "brine" with which the pipes are surrounded. This brine is then circulated by pumps through coils of piping and produces the refrigerating effect. The expanded ammonia-gas is then drawn into the compressor under a suction of from 5 to 20 lbs., thus completing the cycle of operations.

The brine consists of a solution of salt in water. Liverpool salt solution weighing 73 lbs. per cu. ft. (sp. g.=1.17) will not congeal at 0° F. American salt brines of the same proportions congeal at 20° F. Ammonia required=0.3 lb. per foot of piping. Leakage and waste amount to about 2 lb. per year per daily ice capacity of one ton. The brine should be about 6° colder than the space it cools.

Ammonia Absorption System. In this system the compressor is replaced by a vessel,—called the absorber,—where the expanded vapor takes advantage of the property of water or a weak ammoniacal liquor to dissolve ammonia-gas. (At 59° F. water absorbs 727 times its own volume of ammonia-vapor.) The liquor in the absorber is then pumped into a still heated by steam-pipes, where the ammonia-gas is vaporized, the remainder of the process being then the same as in the compression system. The absorption system is less expensive to install, and commercial ammonia hydrate (62% water, sp. g.=0.88) may be used in the absorber.

Efficiency. Ice-melting capacity per lb. of fuel $= wst \div 142.2w_1$; Ice-melting capacity in tons (2,000 lbs.) per day of 24 hours $= 24wst \div (142.2 \times 2{,}000)$, where w=lbs. of brine or other fluid circulated per hour

w_1=lbs. of fuel used per hour, s=specific heat of the circulating fluid, and t=range of temperature experienced by the circulating fluid in degs. F.

Design of a Compression Machine. The weight of the medium required is determined by the condition that each pound must withdraw from the brine the heat necessary to change the liquid medium in the condenser at t (with a heat of liquid in each lb.$=h$) into saturated vapor at t_1 in the vaporizer, where the total heat of evaporation per lb.$=H$. The heat withdrawn per lb. per min., $L=H-h$, and, in ice made per hour, the weight of the medium, $w=142.2\times$lbs. of ice made per hour $\div 60(H-h)$.

Assuming the compression to be adiabatic, the absolute temperature of the superheated vapor leaving the cylinder, $T_s=T_2\left(\frac{p_1}{p_2}\right)^{0.29}$, where T_2 is the absolute temperature (degs. F.) of the vapor in the expansion or vaporizer coils in the brine, and p_1, p_2 are the pressures before and after expansion.

The cooling water required in the condenser, $W=w[k_p(t_s-t_1)+H-h]$ lbs., where k_p=specific heat of the superheated vapor at constant pressure, t_s and t_1=temperatures (F.) of the compression cylinder and condenser respectively, and $(H-h)$=heat of vaporization at the pressure p_1 of condenser.

The H.P. of the steam cylinder driving the compressor

$$=\frac{778w}{33{,}000}[k_p(t_s-t_1)+H_1-H_2],$$

where H_1 and H_2 are the total heats of vaporization at the pressures and temperatures in the condenser and vaporizer, respectively. This value must be increased to allow for heat and friction losses.

The volume of the compressor cylinder $=\frac{w\times\text{vol. of 1 lb. of vapor}}{\text{No. of strokes per min.}}$.

Specific Heats at Constant Pressure (k_p). Ammonia, 0.508; carbonic acid, 0.217; sulphur dioxide, 0.1544.

Temperatures for Cold Storage. Fruits, vegetables, eggs, brewery work, 34° F.; butter, cheese, shell oysters, 33° ; dried fish, canned goods, 35°; flour, 40°. The following should be frozen at the first temperature and then maintained at the second: Butter, 20°, 23°; poultry, 20°, 30°; fresh fish, 25°, 30°; tub oysters, 25°; fresh meat, 25°.

HEATING AND VENTILATION.

Ventilation. Impurities in air are due to carbonic acid and organic particles exhaled from the lungs, water vapor from perspiration, dust, smoke, noxious gases, etc. The measure of impurity, however, is taken as the content of carbonic acid, which should not exceed 6 to 8 parts in 10,000. Fresh air contains 4 parts (country air, 3 to 3.5) in 10,000. The hourly yield of CO_2 per person is 0.6 cu. ft.; consequently each 1,000 cu. ft. of fresh air can take up at least 0.2 cu. ft. of CO_2 and not exceed the limit of 6 parts in 10,000; hence 3,000 cu. ft. of fresh air per person, if uniformly diffused, will keep the respiratory CO_2 down to that limit. It is further found that the atmospheric contents of a room may be changed three times per hour without causing inconvenient draft, hence 1,000 cu. ft. of air space is a proper provision per person. From 2,000 to 2,500 cu. ft. per person per hour is sufficient for auditoriums used but for two or three hours at a time. School-rooms should have at least 1,800 cu. ft. per scholar per hour, and in hospitals from 4,000 to 6,000 cu. ft. per patient per hour should be supplied on account of the various unhealthy excretions.

According to Rietschel (Ing. Taschenbuch) the hourly supply of air per capita in cubic feet should be as follows: Hospitals, adults, 2,600,—children, 1,200; schools, pupils under 10 yrs., 400 to 600,—pupils over 10 yrs., 600 to 1,000; auditoriums, 600 to 1,100; work rooms, 600 to 1,100; living rooms, 1 to 2 times cubic contents; kitchens and closets, 3 to 5 times cubic contents.

Carpenter states that the number of changes of air per hour should be as follows Residences,—halls, 3; living rooms, 2; sleeping rooms, 1. Stores and offices, 1st floor, 2 to 3; upper floors, 1.5 to 2. Assembly rooms, 2 to 2.5.

Heating of Buildings. Let W = sq. ft. of transmitting surface, t_1 = inside temperature, t_2 = outside temperature, both in degs. F. $t = t_1 - t_2$, k = a coefficient representing for various building materials the heat loss by transmission per sq. ft. of surface in B.T.U. per hour for each degree of difference of temperature on the two sides of the material, and H = the total heat transmission = Wkt.

Values of k (Ing. Taschenbuch).

Thickness of wall in inches	4	8	12	16	20	24	28	32	36	40	48
k, for brick	0.53	.38	.30	.25	.22	.19	.17	.15	.13	.12	
Do. sandstone			0.45	.39	.35	.32	.29	.26	.24	.22	.19

For limestone add 10% to values for sandstone.

Solid plaster partitions: 1.75 to 2.25 in. thick, 0.6; 2.5 to 3.25 in., 0.48.

Floors: joists with double floors, 0.07; stone floor on arches, 0.2; planks laid on earth, 0.16; planks on asphalt, 0.2; arch with air-space, 0.09; stones laid on earth, 0.08.

Ceilings: joists with single floors, 0.1; arches with air-space, 0.14.

Windows: single, 1.00; double, 0.46.

Skylights: single, 1.06; double, 0.48.

Doors, 0.4.

The above values should be increased according to conditions as follows: For rooms unusually exposed, add 5%; for N., NE., E., NW. and W. exposures and where height of ceiling (h) exceeds 18 ft., add 10%; for h = 13 ft., add $3\frac{1}{3}$%; for h = 15 ft., add $6\frac{2}{3}$%.

For rooms heated daily, but not at night, add $A = 0.0625\,(N-1)H \div Z$; and for rooms not heated every day, add $B = 0.1(8+Z)H \div Z$, where N = No. of hours between cessation of heating and restarting of fire, and Z = No. of hours from starting of fire until rooms attain required temperature.

In heating assembly rooms account must be taken of the heat given out by audiences and illuminants. A person gives out about 400 B.T.U. per hour, an ordinary gas-burner about 4,800 B.T.U. per hour, and an incandescent electric lamp (16 c. p.) 1,600 B.T.U. per hour. A gas-burner vitiates the air as much as $5\frac{1}{2}$ persons.

B. T. U. per Hour required to Heat a Room. (Carpenter.) No. of B.T.U. $= \left(\frac{nC}{55} + G + \frac{W}{4}\right)t$, where n = No. of changes of air per hour, C = cu. ft. in room, G = sq. ft. of glass, W = sq. ft. of wall surface exposed to outside air, and t = difference between inside and outside temperatures in degs. F.

Radiation. Ordinary bronzed cast-iron direct radiators give out about 250 B.T.U. per hour per sq. ft. of radiating surface, with steam of 3 to 5 lbs. pressure. Unpainted radiating surfaces of the ordinary indirect type give out about 400 B.T.U. per sq. ft. per hour. For hot-water heating 60% of these values may be taken.

Hot-air furnace walls transmit about 500 B.T.U. per sq. ft. per hour if the walls are much extended, and about 800 B.T.U. if the surfaces are smooth, air temperatures at registers being from 100° to 150° F. Boilers when coal-fired will transmit 2,500 to 4,000 B.T.U. per sq. ft. of heating surface per hour, and from 4,000 to 5,000 B.T.U. when coke-fired. Hot-air systems provided with blowers yield transmission values up to 2,000 B.T.U. per sq. ft. per hour.

Approximate Heating Values of Radiating Surfaces. One square foot of radiating surface will heat by direct steam radiation: Dwellings, school-rooms, offices, 60 to 80 cu. ft.; halls, lofts, stores, factories, 75 to 100 cu. ft.; churches, large auditoriums, 150 to 200 cu. ft. For direct high-temperature hot-water heating, take $\frac{5}{6}$ of above values,—for low-temp. hot-water heating, take $\frac{1}{2}$ of same. For indirect radiation, take $\frac{2}{3}$ of the value for direct radiation.

Sizes of Pipes for Steam-Heating. (Wolff.) Allow 0.375 sq. in. sectional area per 100 sq. ft. of radiating surface for exhaust-steam heating, 0.19 sq. in. per 100 sq. ft. when live steam is used, and 0.09 sq. in. per 100 sq. ft. for returns. Each horse-power of boiler capacity will supply from 80 to 120 sq. ft. of radiating surface. ("Steam.") In good hot-water boilers, the ratio between grate area, boiler heating surface, and radiating surface is 1 : 40 . 200.

HYDRAULICS AND HYDRAULIC MACHINERY.

Water (1 part H+8 parts O.)

Degs. F.	Lbs. per cu. ft.	Relative Vol.	Degs. F.	Lbs. per cu. ft.	Relative Vol.
32	62.418	1.00011	100	62.02	1.00686
39.1	62.425	1.00000	120	61.74	1.01138
50	62.41	1.00025	140	61.37	1.01678
60	62.37	1.00092	160	60.98	1.02306
62	62.355	1.00110	180	60.55	1.03023
70	62.31	1.00197	200	60.07	1.03819
80	62.23	1.00332	210	59.82	1.04246
90	62.13	1.00496	212	59.76	1.04332

For sea-water, multiply above weights by 1.026.

Pressure Equivalents.

1 ft. water at 39.1° F. (max. density) = 62.425 lbs. on the sq. ft.,
= 0.4335 lbs. on the sq. in.
= 0.0295 atmospheres on the sq. in.

1 lb. on the sq. ft. at 39.1° F. = 0.01602 ft. of water; 1 lb. per sq. in. = 2.307 ft. of water.

1 atmosphere (29.922 in. mercury) = 33.9 ft. of water.

1 ft. of water at 62° F. (normal temp.) = 62.355 lbs. per sq. ft.
= 0.43302 lbs. per sq. in.

1 inch of water at 62° F. (normal temp.) = 0.036085 lbs. per sq. in.

Hydrostatic Pressure. The pressure of a liquid against any point of any surface upon which it acts is always perpendicular to the surface at that point, and, at any given depth, is equal in all directions and due to the weight of a uniform vertical column of liquid whose horizontal cross-section is equal to the area pressed upon and whose height is the vertical distance from the center of gravity of the surface pressed to the surface of the liquid.

When a liquid pressure is exerted on one side of a plane area, the resultant force experienced by the area is perpendicular to the area, equal to the sum of all the pressures and acts at a definite point called the center of pressure.

Centers of Pressure h (= vertical depth from surface of liquid).

Rectangle: upper side parallel to liquid surface and distance h_1 from same,

$$h=\left(\frac{a}{3}\right)\frac{3h_1+2a}{2h_1+a}+h_1;\quad \text{if}\quad h_1=0,\quad h=\frac{2a}{3}.$$

Triangle: base lying in surface of liquid, $h=a\div 2$; vertex in liquid surface, base horizontal, $h=3a\div 4$.

Circle or Ellipse: $h=a+h_1+\dfrac{a^2}{4(a+h_1)}$; if $h_1=0$, $h=5a\div 4$.

In the above a = vertical height of triangle or rectangle, radius of circle or vertical semi-axis of ellipse.

Buoyancy. When a body is immersed in a liquid it is buoyed up by a force equal to the weight of the liquid it displaces whether floating or sinking. This upward pressure may be considered as acting at the c. of g. of the displaced liquid, or, as it is termed, at the center of buoyancy, and a vert. line drawn through the center is called the axis of flotation. The line connecting the center of buoyancy and the c. of g. of a floating body at rest is called the axis of equilibrium and is vertical. If an external force acting on the body inclines the axis of equilibrium, a vertical line from the center of buoyancy intersects this axis at a point called the metacenter. The equilibrium is stable, indifferent, or unstable, according as the metacenter is above, coincident with or below the center of buoyancy.

Head, Pressure, and Velocity Energy. The pressure of the atmosphere balances the pressure of a column of water 33.9 ft, high. and the "head" of the column, $H = 33.9 \div 14.696 = 2.307p$. If a vertical gauge-tube be inserted in a pipe the water will rise in it to a height proportional to the pressure; then, connecting head and pressure $PA = GHA$, $P = GH$, and $H = P \div G$, where P = supporting pressure in lbs. per sq. ft., H = height of column in ft., G = weight of 1 cu. ft. of water in lbs., and A = area of cross-section of column in sq. ft.

Head and Velocity. A water particle (weight $= w$) at height, H, has a potential energy equal to wH, and when it has fallen through H its kinetic energy $= \frac{wv^2}{2g}$. Neglecting friction and other losses, $wH = wv^2 \div 2g$ and $v = \sqrt{2gH} = 8.02\sqrt{H}$.

Any given portion of water flowing steadily between two reservoirs which are kept at a constant level will,—neglecting friction and viscosity,—possess an unvarying amount of energy which may be due to head, pressure, velocity, or to all three. If a vertical gauge-tube be inserted at any point of the pipe connecting the reservoirs the water will rise in it to a level *below* that of the reservoir from which it flows, a portion of the head energy represented by the difference of levels having become kinetie, and the total head (H_t) eonsists of H due to unexpended fall $+ \frac{P}{G}$ due to pressure (as shown by gauge-tube) $+ \frac{v^2}{2g}$ due to velocity.

Multiplying each by w gives the respective energy, the energy of 1 lb. of water being $H_t = H + \frac{P}{G} + \frac{v^2}{2g}$.

By sufficiently contracting the sectional area of the pipe at some point between the reservoirs the throttling so caused will reduce the pressure below that of the atmosphere and create a partial vacuum. This principle is employed in jet-pumps (efficiencies, 30 to 72%).

Discharge of Water through Orifices. If a reservoir is emptied through an orifice near its bottom, the volume of the water passing, Q = velocity × area of orifice, and, neglecting resistances, The Theoretical Discharge in cu. ft. per sec. $q = Av = 8.02A\sqrt{H}$. On account of resistances v is reduced, and, letting c_1 = coefficient of velocity, $v = 8.02c_1\sqrt{H}$. If the reduced velocity be considered as due to a loss of head, H_r, a coefficient of resistance, ρ, may be adopted, H_r being taken as equal to ρH_1, where H_1 is the remaining or unexpended head. $H = H_1 + H_r = H_1 + \rho H_1 = (1+\rho)H_1$, and $v = 8.02\sqrt{H_1} = 8.02\sqrt{\frac{H}{1+\rho}}$. Also, $c_1\sqrt{H} = \sqrt{\frac{H}{1+\rho}}$, $c_1 = \sqrt{\frac{1}{1+\rho}}$, and $\rho = \frac{1}{c_1^2} - 1$. This loss occurs within the vessel and orifice. A further loss is caused by the contraction of the jet area at a distance from the orifice equal to one-half the jet diam. Let k = coefficient of contraction; then, Actual Discharge in cu.ft. per sec., $q_a = c_1vkA = 8.02kA\sqrt{\frac{H}{1+\rho}}$, or, letting $C = c_1k$ = eoefficient of diseharge, $q_a = 8.02AC\sqrt{H}$.

Average Values of Coefficients.

	Orifices.			
	Sharp-edged.	Re-entrant Cyl.	Cylinder.	Bell-mouthed.
$c_1=$	0.97	1.00	0.82	0.99
$\rho=$	0.0628	0.	0.487	0.02
$k=$	0.64	0.53	1.00	1.00
$C=$	0.62	0.53	0.82	0.99

Measurements of Water-Flow over Weirs. Let a stream be partly dammed and the water allowed to flow through a rectangular notch, or weir, which is beveled to sharp edges on the intake side. To find the discharge, divide the head, H (or distance from edge of notch to surface of water), into small portions, h_1, and consider each small rectangle ($h_1\times$length of notch, L) as a separate orifice. At any depth, H_1, $v=8.02\sqrt{H_1}$ and the discharge through the small rectangle $=8.02L\sqrt{H_1}$. Representing the various discharges by horizontal lines of proportionate length, the figure bounding these lines will be found to be a parabola of base $=8.02L\sqrt{H}$, and height = head H (the lines varying in length as $\sqrt{H_1}$). The total theoretical discharge will then be equal to the area of the parabola, or, $q=\frac{2}{3}\times8.02LH^{\frac{3}{2}}=5.347LH^{\frac{3}{2}}$. The actual discharge is smaller, being, according to the following authorities:

	Both end contractions suppressed.	One suppressed.	Full contraction.
Francis. . .	$q_a=3.33LH^{\frac{3}{2}}$	$3.33\left(L-\frac{H}{10}\right)H^{\frac{3}{2}}$	$3.33(L-0.2H)H^{\frac{3}{2}}$.
Smith. . . .	$q_a=3.29\left(L+\frac{H}{7}\right)H^{\frac{3}{2}}$	$3.29LH^{\frac{3}{2}}$	$3.29\left(L-\frac{H}{10}\right)H^{\frac{3}{2}}$.

(L should not be less than $3H$.)

For flow over a sharp-crested weir without lateral contractions, air being freely admitted behind the falling sheet of water,

$$q_a=\left[0.425+0.21\left(\frac{H}{H_1+H}\right)^2\right]8.02LH^{\frac{3}{2}},$$

where H_1 = height in feet from bottom of channel of approach to the crest of weir (Bazin).

In triangular notches $\frac{L}{H}$ at any depth is constant and therefore C is regular and may be taken as 0.617.

$q_a=\frac{4}{15}CLH^{\frac{3}{2}}\sqrt{2g}=1.32LH^{\frac{3}{2}}$. For a 90° notch, $L=2H$ and $q=2.64H^{\frac{5}{2}}$; for a 60° notch, $L=1.155H$ and $q_a=1.524H^{\frac{5}{2}}$.

The Horse-Power of a Stream $=\dfrac{q_a\times G\times H\text{ (available height of fall)}}{550}$ $=0.1135q_aH$.

Friction in Pipes is independent of the pressure but is proportional to the wetted surface. $F_n\propto Av^2=\mu Av^2$, at moderate velocities, and, as $1.03G=2g$, $F_n=1.03\mu GA\frac{v^2}{2g}$.

If a cylindrical body of water (length L, diam. D) move at a velocity, v, through the pipe, F_n per sq. ft. of sectional area $=1.03\mu G\frac{\pi DL}{0.25\pi D^2}\cdot\frac{v^2}{2g}$ $=4.12\mu\frac{L}{D}\times G\times\frac{v^2}{2g}$, and, as $H=P\div G$, the Head Lost in Friction $=4.12\mu\frac{L}{D}\cdot\frac{v^2}{2g}$.

$\mu=0.004$ for clean, varnished surfaces, 0.0075 to 0.01 for pipes, and 0.009 for surfaces of the roughness of sand-paper.

Wm. Cox's formula: Friction Head $=L(4v^2+5v-2)\div 1{,}000d$, where $d=$ diam. in in. (Pelton Water Wheel Co.).

Flow of Water through Pipes. $v=CR^{\frac{2}{3}}S^{\frac{1}{2}}$. (Tutton.) R (hydraulic radius) = sectional area ÷ wetted perimeter, $=D\div 4$ for round pipes when full or half-full; S (slope) = Head ÷ length of pipe = sine of angle of inclination of pipe. Values of C for various materials: W. I. pipe, 160; new C. I. pipe, 130; used C. I. pipe, 104; lap-riveted pipe, 115; W. I., asphalted, 170; wood-stave pipe, 125; rough, pitted pipe, 30 to 80; brick conduits, 110.

Flow of Water in Open Channels. (Kutter.)

$$v=\sqrt{RS}\left\{\left(41.6+\frac{0.00281}{S}+\frac{1.811}{C}\right)\div\left[1+\frac{C}{\sqrt{R}}\left(41.6+\frac{0.00281}{S}\right)\right]\right\},$$

where S = fall of water surface in any distance ÷ said distance = sine of slope; C = coefficient depending on the character of the channel surface, and having the following values: planed boards, 0.009; neat cement, 0.01; plaster (75% cement), 0.011; rough boards, 0.012; ashlar or brick-work, 0.013; rubble masonry, 0.017; canals, firm gravel, 0.02; canals and rivers in good condition, fairly uniform section, free from stones and weeds, 0.025; same, but with occasional stones and weeds, 0.03; same, in bad condition, many stones and weeds, 0.035; torrents encumbered with detritus, 0.05.

Tutton's formula for pipes may also be used as herewith modified, where C has the values given for Kutter's formula: $v=\frac{1.54}{C}R^{\frac{2}{3}}S^{\frac{1}{2}}$.

Hydraulic Gradient. Water being discharged from a reservoir through a pipe of uniform diameter, the net head at any point may be found by applying a pressure gauge which will show a loss from total head due to velocity, $\frac{v^2}{2g}$ + loss due to friction. The friction loss varying directly as the distance from reservoir, a straight line bounds the heights of the various water columns in the gauges and is called the line of virtual slope, or hydraulic gradient. No part of a pipe should be above this line, as the pressure would then be less than that of the atmosphere and the water would tend to separate.

Loss by Eddies and Shock. Bends, elbows, valves, and cocks produce frictional resistances to flow in systems of piping, which are computed in terms of the head and are to be added to the resistance of the pipe in order to obtain the final discharge.

Water discharged into a basin delivers all of its energy as shock, but whenever a sudden change of velocity takes place eddies are formed which absorb energy. When an abrupt contraction takes place, as from a large pipe to a smaller one, the loss of head $=0.3v_2^2\div 2g$, and for a sudden enlargement of sectional area, loss of head $=(v_1-v_2)^2\div 2g$, where v_1 and v_2 are respectively the velocities in the first and second pipes.

Angles and Elbows. Loss of head $=cv^2\div 2g$. Let β = number of degrees of the angle through which the direction of flow is deviated; then,

for $\beta=$	20	40	60	80	90	100	120	140
$c=$	0.046	0.139	0.364	0.74	0.985	1.26	1.861	2.431

Bends. Loss of head $=c\cdot\frac{\beta}{180}\cdot\frac{v^2}{2g}$. c depends on the ratio of the radius of the pipe ($0.5D$) to the radius of curvature of the bend (R).

$0.5D\div R=$	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.00
$c=$	0.131	0.138	0.158	0.206	0.294	0.44	0.661	0.977	1.408	1.978

Gate-Valves. Loss of head due to partial opening $=cv^2\div 2g$.

Opening =	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$
$c=$	98	17	5.52	2.06	0.81	0.26	0.07

Cocks. Loss of head $=cv^2\div 2g$.

Opening =	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$
$c=$	222	52.6	21.1	7.8	2.8	0.92	0.2

WATER WHEELS.

Pressure on Vanes. Force causing momentum $= \frac{w}{g} f$, and, as $f = v \div t$, $Pt = wv \div g$, or, pressure × time (i.e., impulse exerted) = momentum generated. If $t = 1$ sec. and w = weight of water passing per sec., $wv \div g$ = change of momentum $= P$ = pressure on vane.

Flat Plate or Vane, fixed (its velocity being 0). $P = wv \div g = (GAv)v \div g$.

Flat Plate Moving in the Direction of Jet. (Vel. of plate $= v_2$, vel. of jet $= v_1$.) Water passing per sec. $= GA(v_1 - v_2)$. P = difference of momentum before and after impact, $= [GA(v_1 - v_2)v_1 \div g] - [GA(v_1 - v_2)v_2 \div g] = GA(v_1 - v_2)^2 \div g$.

Moving Hemispherical Surface or Cup. Relative velocity of jet and cup when meeting $= v_1 - v_2$ (forward), and when leaving, $= v_1 - v_2$ (backward). Consequently, the absolute discharge velocity = cup velocity — relative backward velocity, $= v_2 - (v_1 - v_2) = 2v_2 - v_1$, whence, $P = \frac{GA(v_1 - v_2)v_1}{g} - \frac{GA(v_1 - v_2)(2v_2 - v_1)}{g} = \frac{2GA(v_1 - v_2)^2}{g}$. If $v_2 = v_1 \div 2$, the absolute velocity of rejection = 0, and all of the jet energy is exerted on the cup.

Wheel with Radial Vanes, a vane being constantly before the jet: Momentum before impact, $= (GAv_1)v_1 \div g$; after, $= (GAv_1)v_2 \div g$; $\therefore P = \frac{GAv_1(v_1 - v_2)}{g}$.

Wheel with Many Curved Vanes: momentum before impact $= GAv_1^2 \div g$; after, $= GAv_1(2v_2 - v_1) \div g$, $\therefore P = 2GAv_1(v_1 - v_2) \div g$, or twice that of flat radial vanes. In this case and that of the hemispherical cup the direction of the jet water is returned upon itself.

Undershot Wheels are suitable for falls less than 6 feet. Diameter may be 4 × fall. Efficiency, with radial floats or vanes, 30%; with curved floats, about 65%. Circumferential velocity = 55% of the velocity due to head (approx.). As the floats are never filled with water, the action is due to pure impulse, and if the floats are properly curved the water enters without shock and leaves without horizontal velocity. Construction of float curve (Fig. 23): From the center of wheel draw OA vertically and make $AOB = 15°$. Let the jet (of thickness C, $= \frac{1}{6}$ × head) have a slope of 1 in 10. From the middle of jet, D, draw DE so that $ODE = 23°$. Take $DE = 0.5$ to 0.7 × head, and from E strike the arc DF, which is the curve for the Poncelet form of undershot wheel.

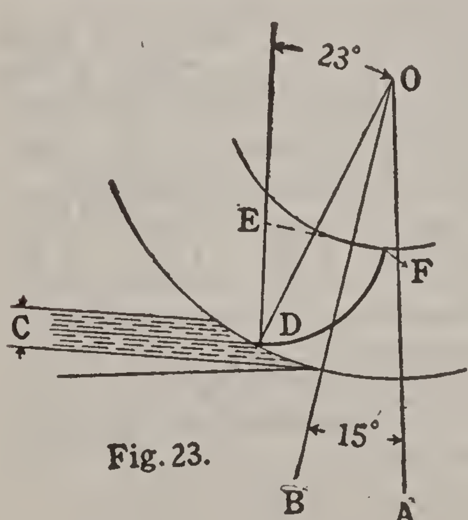


Fig. 23.

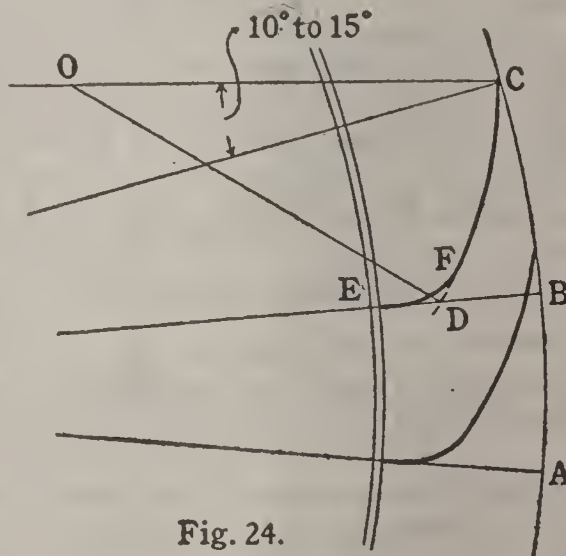


Fig. 24.

Breast Wheels are used for falls from 6 to 12 feet. Efficiency from 60 to 65%. Vanes curved similarly to those of Poncelet wheel.

Overshot Wheels are used for falls ranging from 12 to 70 feet. Efficiency, 70 to 75%. Best circumferential velocity = 6 ft. per sec. = one-half the velocity of the water due to a fall of 2.25 ft.; consequently, point at which water strikes wheel should be 2.25 ft. below the top water level. Construction of float curve (Fig. 24): make $ED = AB \div 3$, and $BC = 1.2AB$. Draw CO 10° to 15° to radius. From O strike the arc FC, F being near to D, and round the arc curve into radial line DE.

The Pelton Wheel is used for heads exceeding 200 feet. In it the water in the form of a jet impinges on a series of cup-shaped buckets affixed to the wheel circumference, to which latter the direction of jet is tangential. These cups are made double, with a center fin which splits the jet and returns the water on the sides, the discharge being effected with but little velocity. Efficiency, from 80 to 90%. Bucket velocity should be one-half jet velocity.

TURBINES.

Turbines are water wheels in which the motion is caused by the reaction of the water pressure between stationary guide blades and the vanes or floats of the wheel. The water flow may be axial or radial (inward or outward) in direction, and it should be so deviated that it enters the wheel floats as nearly at a tangent as possible, and leaves either radially or in a direction parallel to the axis as the case may be.

Radial Outward-Flow Turbines (Fourneyron type). Q = cu. ft. water passing per sec. under a head of H feet. Inner radius $R_1 = 0.326\sqrt{Q}$; outer radius $R = cR_1$, where $c = 1.25$ to 1.5. Angle of guide at entrance $\alpha = 15°$ to 30°. Angle of bucket at same point, $\beta = 2\alpha + 20°$ to 30°. The

velocity of wheel at $R_1 = v_1 = \sqrt{\dfrac{2gH}{\dfrac{2\sin\beta\cos\alpha}{\sin(\beta-\alpha)} + 0.1\left[\left(\dfrac{\sin\beta}{\sin(\beta-\alpha)}\right)^2 + c^2\right]}}$.

(If $\alpha = 15°$, $\beta = 60°$, $c = 1.5$, $v_1 = 4.84\sqrt{H}$.)

Velocity at $R = v = cv_1$; r.p.m. $= 60v \div 2\pi R = 9.55v \div R$.

Velocity through guide passages, $v_2 = v_1 \sin\beta \div \sin(\alpha - \beta)$. Area of cross-section of all openings $= Q \div v_2 = A = Q \sin(\alpha-\beta) \div v_1 \sin\beta$.

If D = depth and B = width of a bucket, $D \div B = \lambda = 2$ to 5, inversely according to the head of water. Thickness of metal floats, $T = 0.015R$.

$D = \dfrac{A}{2\pi R_1 \sin\alpha}\left[1 + \left(\dfrac{2\pi R\lambda T \sin\alpha}{A}\right)\right]$. Number of guides, $N_1 = \lambda A \div D^2$. No. of wheel buckets, $N = N_1 \sin\beta \div \sin\alpha$. Angle of discharge, δ: $\sin\delta = (A_1 + NTD) \div 2\pi RD$, where A_1 = area of discharge openings.

Curvature of floats (Fig. 25): Draw $CAB = \delta$, drop CB perpendicular to AB. $AD = AE = B \div 2$. Set off BF and $BG = AD$. From F strike the arc HD. Draw $DK = CL$, making $BDK = 180° - \beta$, and join CK. Bisect CK at M and draw the perpendicular MN. Draw arc DL from N as a center. Draw PL and CP, each inclined to CL by $\alpha°$. From P as a center strike the arc RL of guide blade. Inward-flow turbines are designed similarly, but in an inverse manner.

Axial or Parallel-Flow Turbines (Jonval type). The guide blades in this type are arranged in the form of a ring above the wheel vanes, the water flowing parallel to the axis. These wheels work best when submerged in the tail-race or connected thereto by a draft-tube whereby the suction of the latter may be availed of. $\alpha = 15°$ to 25°, $\beta = 100°$ to 120°. Velocity, v, same as velocity v_1 of the Fourneyron wheel where $c = 1$. Velocity of entering water $= v_1 = v \sin\beta \div \sin(\beta - \alpha)$. Total sectional area of entrances between guides, $A = Q \div v_1$; total discharge area, $A_1 = Q \div v$. Mean radius, $R = (R_1 + R_2) \div 2$; radial width of operative ring of wheel, $D = R_2 - R_1 = 0.4R$, and $R_1 = 0.8R$; $R_2 = 1.2R$. $\lambda = D \div B = 2$ to 4.

R, approx. $= \sqrt{\dfrac{A}{0.8\pi\sin\alpha}}$; $T = 0.02R$. No. of guides, $N_1 = (A \div BD) + (\lambda A \div D^2)$; No. of floats, $N = N_1 \sin\beta \div \sin\alpha$. $\sin\delta = (A_1 + NTD) \div 2\pi RD$. R.p.m. $= 9.55v \div R$. Height of wheel $= (0.5$ to $0.6)R$.

Curvature of floats (Fig. 26): Both the guides and floats are warped surfaces generated by a line at right angles to the axis, whose outer end

follows the curves in the figure. Draw AB inclined to the plane of wheel by $\alpha°$, and similarly DC at $\delta°$. Draw BF perp. to AB. From F as a cen-

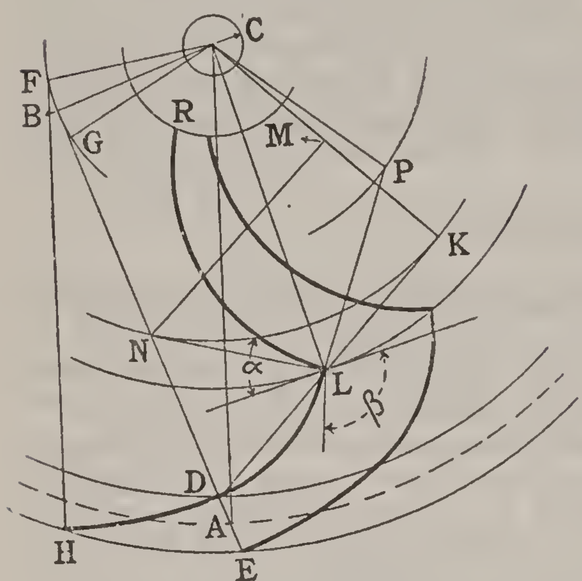


Fig. 25.

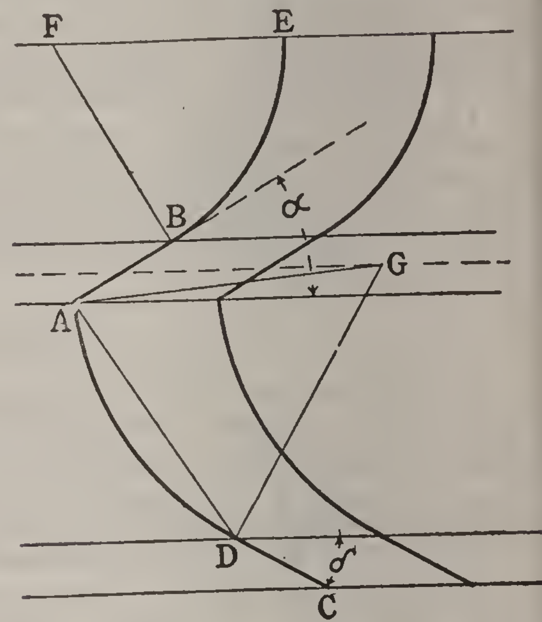


Fig. 26.

ter, strike the arc BE. Draw DG perp. to DC, make angles $GDA = DAG = (\beta+\delta) \div 2$, and from intersection G, as a center, draw arc DA. The lower parts of guide and float (AB and CD) are straight lines.

Impulse Turbines (Girard type) are parallel-flow wheels with the wheel passages so enlarged toward the outlet and ventilated that they are never entirely filled with water, the energy being purely due to velocity. They are regulated by entirely closing a number of the guide passages, the efficiency (60 to 80%) being therefore unimpaired by fractional opening.

Modern Practice. (From articles by J. W. Thurso in E. N., Dec., '02.) For heads less than 20 ft., use radial-inflow reaction (Francis) turbines with vertical shafts; for heads of 20 to 300 ft., the same, but with horizontal shafts; for heads exceeding 300 ft., use radial, outward-flow, free-deviation turbines with horizontal shafts, or Pelton wheels.

Parallel-flow turbines are now largely abandoned on account of their poor regulating qualities. Free deviation may be obtained with an efficiency of 70% at 0.2 gate, and 80% at full gate; reaction turbines with 60% efficiency at 0.2 gate and 78% at full gate. (Highest eff., 80% between 0.8 and 0.9 gate.)

Reaction wheels are either regulated by making the guide-vanes movable, so that the openings may be reduced according to load and without materially altering the direction of flow, or, the guide and wheel vanes are divided by crowns into three or more superposed turbines, any number of which may be shut off by a cylindrical gate according to load, allowing those in operation to work at full gate and at the correspondingly higher efficiency.

Free-deviation turbines to attain high efficiencies must work in the free air, and, in order to obtain the advantages of draft-tubes, they must be supplied with air-valves which will automatically keep the water-level below and clear of the wheel.

Draft-Tubes. The use of draft-tubes permits turbines to be mounted on horizontal shafts and also to be set above the tail-water without loss of a part of the head. The hanging water-column in the draft-tube is balanced by atmospheric pressure and could theoretically attain a height of 34 ft. if the water were at rest,—but, with the water in motion, it cannot exceed $\left(34-\frac{v^2}{2g}\right)$ ft., where v = velocity of water in ft. per sec. When

leaving the draft-tube v should not be less than 2 ft per sec. when starting at full capacity, not less than 3 ft. per sec. for variable loads over half capacity, and from 4 to 6 ft per sec. for widely fluctuating loads at times of small capacity. The absolute velocity of the water issuing from wheel in ft. per sec., $v_1=c\sqrt{2gH}$, where $c=0.285$ for large turbines and low leads (10 ft.), 0.2 for medium turbines and heads (100 ft.), and 0.167 for small turbines and high heads (500 ft) H=total head in feet.

When $H=10$ ft., $v_1=7.23$ ft per sec The head, h_1 (due to velocity v_1) $=7.23^2\div 2g=0.812$ ft. Let v be the velocity at which water leaves the draft-tube=3 ft.; the corresponding velocity head, $h=3^2\div 2g=0.14$ ft.; the gain in head by using draft-tube $=h_1-h=0.812-0.14=0.672$ ft., or 6.72% of H. 75% of this gain should be realized in practice.

Under average conditions, the greatest draft head, H permissible for various diameters D of draft-tubes is as follows $D=0.5$ ft., $H=32.5$ ft.; $D=8$ ft., $H=14.5$ ft.; $D=9$ ft., $H=13$ ft.; $D=13$ ft., $H=10$ ft. From these heads should be deducted $h\left(=\frac{v^2}{2g}\right)$ due to velocity, v, of water leaving tube. Short draft-tubes of small diam. should extend from 6 to 12 in. below surface of tail-water,—long tubes of large diam. from 20 to 24 in. below. Tubes should have a gradual taper, enlarging towards the tail-water, in order to reduce the velocity of the discharge and to thus avoid shock.

The H. P. of a Water Wheel $=GqH\eta\div 550$, where η=efficiency of wheel. As the water has no forward momentum on leaving the turbine (or on entering a centrifugal pump), each lb. undergoes a change of momentum $=v\div g$, where v is the forward component of the entering velocity (leaving vel. for centrifugal pump). Let v_1=velocity of wheel-rim; then, useful work per lb. water $=(vv_1\div g)$ ft.-lbs. per sec. $=\eta H$.

High-Efficiency Turbines. Samson (Leffel) and McCormick (S. Morgan Smith & Co.) turbines tested at the Holyoke flume under heads of about 15 ft. show efficiencies of over 80% at full and ⅘ gate, and a maximum of about 85% at ⅞ gate.

Losses in Turbines. Surface friction and eddying, 10 to 14%; energy rejected into tail-race, 3 to 7%; shaft friction, 2 to 3%.

PUMPS.

Centrifugal Pumps are simply reversed turbines in which the application of mechanical power to the wheel transforms velocity into pressure and elevates water to the same height (neglecting losses) as the head would be for a turbine running at the same speed. The radial outward-flow type is best adapted for pumping. Water may be raised through suction up to 26 ft., and, using as a force-pump, may be elevated upwards of 100 ft. by well-designed wheels.

It is claimed for the Worthington volute type that it will work up to heads of 85 ft., and that tests have shown an efficiency of 86%.

A Swiss pump (Sulzer Bros., wheel diam. of 20 in., 890 r.p.m.) tested in 1902, lifted 1,010 gal. (135 cu. ft.) per min. against a head of 428 ft., or, as pump was four-stage, 107 ft. head per wheel. Efficiency, 76%.

A single-stage De Laval pump (runner diam. of 13.75 in., 1,545 r.p.m.) driven by a 55-H.P. steam turbine of same make (tested by Profs. Denton and Kent in Apr. '04) lifted 1,760 gal. per min. 100 ft. with an efficiency of 75%. Duty, condensing, 61,860,000 ft.-lbs. per 1,000 lbs. of commercially dry steam (moisture $<1.7\%$) and 45,000,000 ft.-lbs. per 1,000 lbs. steam (non-condensing).

A two-stage pump of same make (runners 9 in. and 2.84 in. diam., 2,050 and 20,500 r.p.m. respectively) lifted 244 gal. per min. 781 ft. Duty, 48,880,000 ft.-lbs. per 1,000 lbs. steam. Steam per water-H.P. (lbs. water lifted per sec. × lift in ft. ÷ 550) = 40.5 lbs. per hour.

Proportions. Let R=wheel radius, and $R_1=\frac{R}{2}$=radius of water inlet. Diam. of discharge-pipe, $D=0.36\sqrt{Q\div\sqrt{2gH}}$. Diam. of wheel $=2R$ $=0.18\sqrt{Q\div\sqrt{H}}$. To draw curve of wheel-vane: Let v_1=velocity of inflowing water. Draw radius R; at distance R_1 on this radius draw a

line inclined outward to R by angle α, whose tangent $=0.0176NR \cdot \sqrt{v}$ (N = r.p.m. Q = cu. ft. per min. v_1 = ft. per sec.).

The vane curve must be tangential to this line. At the extremity of a radius draw a tangent and on this tangent, at a point distant $l \left(=\frac{R^2 - R_1{}^3}{2R_1 \sin \alpha}\right)$ as a center, strike an arc from the outer circumference of wheel to the inlet circumference, and this arc will be the vane curve.

The case should start at zero cross-section and increase in one circumference to full discharge section by means of an Archimedean spiral.

Hydraulic Ram. Water flowing in a pipe under a low head escapes through an opening at the end until it acquires a velocity sufficient to move a valve closing the outlet. This sudden stopping of flow creates an excessive pressure in the pipe, and a valve near the end is opened which leads to an air-chamber into which the water rushes, and from there into a delivery-pipe. Equilibrium being restored the air-chamber valve closes, outlet valve opens and the cycle is repeated. Water may be raised 10 times as high as the head of the stream in ft. Efficiency, 50 to 75%.

Pulsometer. In this device water is raised by suction into the pump chamber by a vacuum resulting from the condensation of steam within it; it is then forced into the delivery pipe by the pressure of a fresh supply of steam. Two chambers are employed, one raising while the other discharges. Duty, 10,000,000 to 20,000,000 ft.-lbs. per 1,000 lbs. of steam.

The Air-Lift Pump. A vertical pipe with its lower end submerged in a well or tank is supplied with a smaller pipe from which compressed air enters into the bottom of the larger pipe.

The column of liquid in the pipe, consisting to a certain extent of air-bubbles, is lighter than an equally high column of liquid not so aerated, and therefore rises. The efficiency ranges from 25 to 50%, where the ratio of submerged length to length above surface varies from 0.5 to 2, respectively. As there are no moving parts, this device is valuable in the case of lifting acids, chemical solutions, sewage, etc.

PLUNGER PUMPS AND PUMPING ENGINES.

Quantity of Water Pumped. Q (in cu. ft. per min.) $=0.00545Vd^2$; Q_1 (gals. per min.) $=0.040766Vd^2$, where V = speed of plunger in ft. per min. and d = diam. of plunger in in. V ranges from 100 to 200 ft. per min., and in well-designed engines may reach 250 ft. if the waterways are ample and the water is not abruptly deflected. Loss by leakage and slip ranges from 5% for new, well-packed pumps to 40% for worn and badly cared-for apparatus

H. P. Required to Raise Water a Given Height, H. (Theoretical.) H.P. $=QH \div 529.2 = Q_1H \div 3{,}958.7$, or, as 1 ft. $H=2.31$ lb. pressure, p, H.P. $=Qp \div 229.2 = Q_1p \div 1{,}714.5$. Theoretical lift for normal temperatures $=34$ ft. When the temperature of the water increases, the pressure of the water vapor decreases the theoretical lift, which at 150° F. = 25.7 ft., at 175° F. = 18.5 ft., and at 200° F. = 7.2 ft. Hot water should therefore flow to the pump by gravity.

Air-Chambers. Even flow and smooth running are obtained by the use of air-chambers, where the impact of the water is received and given out as pressure. On the delivery side these should be from 3 to 6 times the capacity of pump, and on the suction side from 2 to 3 times the capacity.

High-Duty Pumping Engines. Small pumps are either driven from a crank-shaft or are direct-acting, i.e., having a steam cylinder in which the full pressure of the steam is used throughout the stroke. In large, high-duty engines the steam is used expansively.

In the Worthington high-duty engines compensating cylinders are employed in order to equalize the driving force. These cylinders rock on trunnions, are connected to an accumulator under a water pressure of about 200 lbs. per sq. in., and have their plungers pivoted to the pump-rod. This arrangement offers a resistance to the steam pressure during the early part of the stroke, receiving energy during the period of full steam pressure and giving it out later when the pressure falls through expansion, thus maintaining a fairly even effective pressure throughout the stroke

Duty. The old measure of pumping-engine performance was the number of ft.-lbs. of work done per 100 lbs. of coal consumed. In 1891 the A. S. M. E

committee recommended that it be changed to the number of ft.-lbs. of work per million heat units furnished to the boiler (=100 lbs. coal where each lb. imparts 10,000 heat units, or where the evaporation from and at 212° F.=10.355 lbs. water per lb. of fuel). It is customary now to also state the duty in terms of the number of ft.-lbs. of work per 1,000 lbs. of steam used.

Performance of a Modern Pumping Plant. The following data are taken from a 24-hour duty trial of one of the units of the Central Park Ave. pumping plant in Chicago (E. N., 5-26-04), and will serve as an illustration of high-grade installations.

Three Worthington high-duty, triple-expansion engines make up the plant, each with a rated capacity of 20,000,000 gals. per 24 hours against 150 ft. head. Cylinders are 21, 33, and 60 in. in diam., 50 in. stroke, steam-jacketed all over. Superheated steam is used which is supplied by six 225 H.P. Scotch marine boilers, each with two 40 in. corrugated Morison furnaces and 140 2½ in. tubes. Boilers are 10 ft. in diam. and 12 ft. long, fitted with Hawley down-draft furnaces.

Steam pressure at throttle, h.p. and i.p. jackets and reheater coils, 114.45 lbs.; at l.p. jacket, 10.13 lbs. Vacuum in exhaust, near l.p. cyl.= 26.98 in. of mercury, barometer, 14.45 lbs. (The weights of pistons, plungers, etc., are exactly balanced by a water pressure of 78.97 lbs.) Delivery pressure of water=52.23 lbs.=120.65 ft. head. Height of delivery gauge above water=32.24 ft. ∴ Total head=152.89 ft. Temp of water=72° F., temp. of feed-water=102.18° F., temp. of steam at throttle= 516.91° F. (superheated 154°) Total steam used in cylinders=143,734 lbs. Steam used in jackets and reheater, 16,400 lbs. Total steam used, 160,134 lbs. Dry coal burnt to evaporate total steam, 18,534 lbs. R.p.m., 19.33. Piston speed, 159.74 ft. per min. Stroke, 49.587 in. Plunger displacement (24 hrs.), 22,086,318 gals.=2,952,400 cu. ft.=183,934,538 lbs. Allowance for leakage and slip, 0.5%. Net work (24 hrs.), 27,981,142,800 ft.-lbs. Net delivered H.P =588.82. I.H.P.=660.9. Efficiency, 89.15%. Steam per I.H.P. per hr., 10.01 lb.; do., per net delivered H.P , 11.32 lb. Dry coal per I.H.P. per hr., 1.42 lb.; do., per net delivered H.P., 1.58 lb. Combustible per I.H.P. per hr., 1.07 lb.; do., per net delivered H.P., 1.2 lb. Duty: per 1,000 lbs. steam=174,735,801 ft.-lbs. Duty per 100 lbs. coal= 150,971,958 ft.-lbs.

Boilers Fuel, Maryland Smokeless coal. Upper grate surface, 35 sq. ft. Water heating surface, 1,402 sq. ft. Superheating surface: internal, 180 sq. ft., external, 375 sq. ft. Total coal burnt, 22,779 lbs. Per cent moisture, 0.88. Total dry coal, 22,519 lbs. Per cent ash and refuse, 8.17. Total water fed to boiler, 195,153 lbs. Factor of evaporation (including superheat), 1.166. Equivalent water evaporated into superheated steam from and at 212°, 227,548 lbs. Dry coal per hour per sq. ft. of upper grate surface, 26.87 lbs. Equivalent evaporation from and at 212° per sq. ft. of heating surface, 6.7 lbs. Average steam pressure, 154.22 lbs. Temp. of feed-water entering purifier, 177.26° F. Temp. of escaping gases, 459° F. Degrees of superheat, 162. H.P. developed, 275. Actual water evaporated per lb. of coal fired, 8.567 lbs. Equivalent evaporation from and at 212° F.: of coal fired, 10.077 lbs.; of dry coal, 10.11 lbs.; of combustible, 10.97 lbs. Calorific value of dry coal per lb., 14,213 B.T.U.; do. of combustible, 15,634 B.T.U. Efficiency of boiler (based on combustible), 67.76%; do., including grate (based on dry coal), 64.52%. Cost of coal per ton of 2,000 lbs., $2.89. Cost of coal to evaporate 1,000 lbs. water from and at 212° F., $0.151. A similar engine at 142.27 lbs. steam pressure, 71.2° superheat gave a duty of 157,133,000 ft.-lbs. per 1,000 lbs. steam used.

The highest recorded duty (181,068,605 ft.-lbs. per 1,000 lbs. dry steam) is that of an Allis triple-expansion pumping engine at St. Louis, operating under 140 lbs. steam pressure. Another high-duty engine is a Reynolds triple-expansion vertical engine at Boston, 30,000,000 gals. capacity, operating at a piston speed of 195 ft. per min. under 185 lbs. steam pressure. Duty, 178,497,000 ft.-lbs. per 1,000 lbs. steam, or, 163,925,300 ft.-lbs. per million heat units. B.T.U. per I.H.P. per min.=196. Steam per I.H.P. hour=10.375 lbs. Coal per I.H.P. hour=1.06 lbs. Thermal efficiency, 21.63%, or, including economizer, 22.58%.

HYDRAULIC POWER TRANSMISSION.

Water under high pressures (600 to 2,000 lbs. per sq. in.) is advantageously used where power distribution is desired over small areas, viz., wharves, boiler and bridge shops, for presses, cranes, riveting, flanging and forging machinery. The system consists of pumps to develop the desired pressure, from which the water flows through piping to an accumulator, which is a vertical cylinder provided with a heavily weighted plunger. Pipes lead from the accumulator to the machines to be operated. The work stored in an accumulator is equal to the weight on plunger × height in ft. plunger is raised, or wH ft.-lbs. Accumulator efficiency may be 98%. Efficiency of a direct plunger or ram in a hydraulic crane is around 93%, decreasing in proportion to the number of multiplications of movement by pulleys. (Pressures used in boiler shops range from 1,500 to 1,700 lbs. per sq. in.) Effective pressure (lbs. per sq. in.) = accumulator pressure (lbs. per sq. in.) $\times (0.84 - 0.02\ m)$, where m = ratio of multiplying power (H. Adams).

Maximum hoisting speeds in ft. per sec.. warehouse cranes, 6; platform cranes, 4; passenger and wagon hoists, heavy loads, 2; plunger passenger elevators, direct stroke, 10.

Cast iron should not be used for hydraulic cylinders when pressures over 2,000 lbs. per sq. in. are used, W. I. or steel being substituted. The test pressure should be about three times the working pressure.

Design of Hydraulic Cylinders. (Kleinhans.) Load on ram, in tons $= 0.0003927pd^2$; thickness of walls of cylinder in in. $= pD \div 2(f - p)$; thickness of bottom end of cylinder at center $= 0.5D\sqrt{p \div f}$; thickness (at a radius $D \div 3$) between center and wall diam. $= 0.433D\sqrt{p \div f}$; where p = water pressure in lbs. per sq. in., d = diam. of ram or plunger, D = internal diam. of cylinder $= d + 1$ to 2 in., according to size, f = safe fiber stress = 10,000 for cast steel. The bottom of cylinder is spherical (of radius d) and rounded to wall of cylinder by a radius $= 0.2d$.

Friction of Cup Leathers. F = frictional resistance of a leather in lbs. per sq. in. of water pressure $= 0.08p + (c \div d)$, where d = diam. of plunger in in., p = water pressure in lbs. per sq. in., and $c = 100$ for leathers in good condition, 250 if in bad condition. (Goodman.)

SHOP DATA.

THE FOUNDRY.

Sand. Good, new sand contains from 93 to 95% of silica, 5% of alumina, and traces of magnesia and oxide of iron. Sand containing lime should not be used. Floor sand: old sand, 12; new sand, 4; coal dust, 1. Facing sand: old sand, 6; new sand, 4; coal dust, 1. (The numbers refer to parts by weight.)

Loam is a mixture of clay, rock sand, powdered charcoal, cow hair, chaff, horse manure, etc. (for binding power and porosity) ground together in a mill.

Cores require a mixture of rock sand and sea sand with a binding substance, and are black-washed after baking with a mixture of powdered charcoal and clay water.

Parting Sand. Powdered blast-furnace slag, brick dust or fine dust from castings may be used for this purpose. Plumbago, powdered charcoal, soapstone, and French chalk are used for facing moulds in order that smooth castings may be obtained.

Consistency of Sand. If too much burnt, or old sand is used it will cake in the mould. Sand should be so moistened that if the hand is closed on a ball of same and then opened, the sand will just retain the shape given to it.

Shrinkage of Castings. Patterns having one horizontal dimension under 3 in. should be made $\frac{1}{32}$ in. smaller to allow for rapping. Under ordinary conditions the shrinkage of castings per foot is as follows: cast and malleable iron, $\frac{1}{8}$ in.; brass, aluminum, and steel, $\frac{3}{16}$ in.; zinc, $\frac{5}{16}$ in.; tin, $\frac{1}{12}$ in.; white metal, $\frac{1}{32}$ in.; gun-metal, $\frac{1}{8}$ in. The edges of patterns should be rounded, all corners and angles being filleted in order to avoid the weakening due to crystallization in cooling.

Weights of Castings. Multiply weight of pattern by 12.5, 14.1, or 16.7, respectively, if the pattern is of red, yellow, or white pine and the casting is of iron. If the casting is of yellow brass, multiply similarly by 14.2, 16, or 19.

To Clean and Brighten Brass Castings. In a glazed vessel mix 3 parts of sulphuric acid with 2 parts of nitric acid and add a handful table salt to each quart of the mixture. Dip the castings in the mixture and then thoroughly rinse in water.

The Cupola. Speed of melting: $W=2d^2\sqrt{p}$. Air required $Q=0.5d^2\sqrt{p}$. H.P. to operate fan$=d^2\sqrt{p}\div 3{,}800$. In these formulas d=inside diam. of cupola lining in in., W=lbs. of iron per hour, p=air pressure at cupola in ounces per sq. in., and Q=cu. ft. of air per min. (E. N., 7-21-'04).

THE BLACKSMITH SHOP.

Welding. Wrought iron welds at a white, sparking heat (1,500° to 1,600° F.), sand being used as a flux and to prevent scale. Steel welds at lower heats, borax being the flux employed.

Electric Welding. Extra sound welds can be made by abutting the surfaces of the parts to be welded, allowing an electric current of large volume to flow, and by forcing the parts together when the localized heat at the joint (due to the current) has attained the welding temperature. Alternating currents of low potential are used. In general, from 25 to 30 H.P. applied to the generator are required per sq. in. of section to be welded. For iron and steel this power must be applied for [(area in

sq. in.×18)+10] seconds. Copper requires 82 H.P. per sq. in. of section, and it must be applied [(area in sq. in.×17.5)+7] seconds.

To Anneal Tool Steel, heat to an even red and cool slowly in a box, surrounding the steel by gravel and charcoal.

Case-Hardening. Raise the pieces (W. I. or mild steel) to a red heat and apply equal parts of prussiate of potash and salt. Quench while the mixture is flowing, not waiting until it burns off. If extreme hardness is desired, use cyanide of potassium. (A dangerous poison.)

Tempering of Steel. Harden by heating to a dark red (about 1,300° F.), cooling quickly in water, the article being kept in motion. To temper, brighten the surface of the article and heat slowly (not in contact with the flame) until the desired color (as below) appears, and then quench in water or oil.

Very pale straw (430° F.), for brass scrapers, hammer faces, lathe and planer tools for steel and ivory, and bone-working tools.

Light straw (450° F.), for drills, milling cutters, lathe and planer tools for iron.

Medium straw (470° F.), for boring cutters.

Very dark straw (490° F.), for taps, dies, leather-cutting tools.

Brown-yellow (500° F.), for reamers, punches and dies, gouges, stone-cutting tools.

Yellow-purple (520° F.), for flat drills for brass, twist drills, planes.

Light purple (530° F.), for augers, dental and surgical instruments.

Dark purple (550° F.), for cold-chisels, axes.

Dark blue (570° F.), for springs, screw-drivers, circular saws for metal, wood-chisels, wood-saws, planer knives and moulding cutters.

Forgings. Allowance for Machining.

Diam.	up to 5 in.	6 to 8 in.	9 to 10 in.	12 in. and larger
Allowance	0.25 in.	0.375 in.	0. 5 in.	1 in.

THE MACHINE SHOP.

Punches and Dies. Diam. of hole in die = diam. of punch + (0.16 to 0.3) × thickness of plate to be punched, according to various authorities. A fair average value for the excess is 0.2 × thickness.

Cutting Speeds for Lathes, Planers, and Shapers in ft. per min. (Ordinary tool steel.)

	American Practice. (J. Rose.)	German. (Ing. Taschenbuch.)
Hard cast steel		6 to 10
Tool steel	12	12
Machinery steel	15 to 20	18 to 30
Wrought iron	18 " 35	18 " 30
Cast iron	20 " 38	16 " 24
Bronze	60 " 120	40 " 90
Copper	150 " 350	40 " 90

Circumferential speed, ft. per min. = 0.2618 × r.p.m. × diam. of piece in in. Planer speeds range from 18 to 22 ft. per min. Maximum Feeds and Depth of Cuts (Ing. Taschenbuch): max. feed per rev. = 0.06 in. for roughing, and 0.2 in. for finishing; greatest depth of cut = 0.4 in. for C. I., = 0.28 in. for W. I., = 0.16 in. for steel, = 0.12 in. for bronze. Max. planer feed per stroke = 0.08 to 0.16 in. for roughing, and 0.12 to 0.5 in. for finishing; greatest depth of planer cut = 0.8 in. for C. I., = 0.5 in. for W. I., = 0.32 in. for steel, = 0.16 in. for bronze.

Milling Cutters. (Ordinary tool-steel.) Angle of tooth: Front face radial; tooth angle, 50°; angle at cutting edge = 85° (5% clearance). No. of teeth = 2.8 (diam. in in. + 2.6 in.). Take nearest even number.

	Speed, ft. per min.	Depth of cut, in.	Feed, in. per min.
Hard steel	21	$\frac{1}{32}$	$\frac{3}{4}$
Wrought iron	40	1	$\frac{5}{8}$
Mild steel	30	$\frac{1}{4}$	$\frac{3}{4}$
Gun-metal	80	$\frac{1}{2}$	$\frac{3}{4}$
Cast-iron gears	26	$\frac{1}{2}$	$\frac{3}{4}$
Hard cast iron	30	$2\frac{1}{2}$	$\frac{5}{16}$

For light cuts, speed in ft. per min.: steel, 45; W. I., 60; C. I., 90; gun-metal, 105; brass, 120. For heavy cuts reduce these speeds about one-half.

Twist Drills (of ordinary tool-steel). Revs. per min for iron: $\frac{1}{8}$ in., 660; $\frac{1}{4}$ in., 320; $\frac{3}{8}$ in., 220; $\frac{1}{2}$ in., 160; $\frac{5}{8}$ in., 130; $\frac{3}{4}$ in., 105; 1 in., 80; $1\frac{1}{2}$ in., 54; 2 in., 39; 3 in., 26; 4 in., 17. For steel take 0.7 of these speeds,—for brass, multiply them by 1.25.

Feed:—125 revs. per inch depth of hole for drills under $\frac{1}{2}$ in.; for larger drills allow 1 in. of feed per min.

Morse Standard Tapers for Drill Shanks and Sockets.

No. of taper.	Large diam. of socket.	Diam. $\frac{1}{16}$ in. from bottom of hole.	Depth of hole.	C. to c. of slot drill-hole.	Width of slot.	Diam. of tongue.	Length of tongue.
1	0.475	0.369	$2\frac{3}{16}$	$\frac{5}{8}$	0.213	0.33	$\frac{5}{16}$
2	0.7	0.572	$2\frac{5}{8}$	$\frac{3}{4}$	0.26	$\frac{17}{32}$	$\frac{3}{8}$
3	0.938	0.778	$3\frac{1}{4}$	$\frac{7}{8}$	0.322	$\frac{3}{4}$	$\frac{7}{16}$
4	1.231	1.026	$4\frac{1}{8}$	$\frac{15}{16}$	0.478	$\frac{31}{32}$	$\frac{1}{2}$
5	1.748	1.475	$5\frac{1}{4}$	$1\frac{1}{8}$	0.635	$1\frac{13}{32}$	$\frac{5}{8}$
6	2.494	2.116	$7\frac{3}{8}$	$1\frac{3}{8}$	0.76	2	$\frac{7}{8}$

The tongues of drills are 0.01 in. less in thickness than the width of slot. Keys to force out drills are tapered 1.75 in 12 (or 8° 19′).

Taper Turning. Distance tail-center is to be set over=

$$\frac{\text{total length of piece}}{\text{length of tapered part}} \times \frac{\text{diff. between diams. at ends of taper}}{2}.$$

As the centers enter the work an indefinite distance, this rule is only approximate and the results must be corrected by trial.

Machine Screws.

Wire gauge.	Threads per in.	Diam. in in.	Tap drill.	Wire gauge.	Threads per in.	Diam. in in.	Tap drill.
2	56	0.0842	No. 49	12	24	0.2158	No. 17
3	48	.0973	45	14	20	.2421	13
4	36	.1105	42	16	18	.2684	6
5	36	.1236	38	18	18	.2947	1
6	32	.1368	35	20	16	.3210	$\frac{1}{4}$ in.
7	32	.1500	30	22	16	.3474	$\frac{9}{32}$ "
8	32	.1631	29	24	16	.3737	$\frac{19}{64}$ "
9	30	.1763	27	26	16	.4000	$\frac{21}{64}$ "
10	24	.1894	25	28	14	.4263	$\frac{11}{32}$ "
				30	14	.4520	$\frac{23}{64}$ "

Maximum lengths: No. 2, $\frac{1}{2}$ in.; No. 4, $\frac{3}{4}$ in.; No. 6, 1 in.; No. 8, $1\frac{1}{4}$ in.; No. 10. $1\frac{1}{2}$ in.; No. 14, 2 in.; No. 18, $2\frac{1}{2}$ in.; No. 22 and larger, 3 in. Lengths increase by 16ths up to $\frac{1}{2}$ in., by 8ths from $\frac{1}{2}$ to $1\frac{1}{2}$ in., and by 4ths above $1\frac{1}{2}$ in.

International Standard Threads (Metric). Angle of thread=60°; flat $\frac{1}{8}$ ht. of sharp V thread; root filled in $\frac{1}{16}$ ht. Dimensions in mm.

Diam.	Pitch.	Diam.	Pitch.	Diam.	Pitch.
6 & 7	1	18, 20 & 22	2.5	48 & 52	5
8 & 9	1.25	24 & 27	3	56 & 60	5.5
10 & 11	1.5	30 & 33	3.5	64 & 68	6
12	1.75	36 & 39	4	72 & 76	6.5
14 & 16	2	42 & 45	4.5	80	7

Metric threads may be cut in lathes whose lead-screws are in inch pitch by introducing change gears of 50 and 127 teeth. (127 cm.=50 in., within 0.0001 in. For less accurate work a 63-tooth wheel will give an error of only 0.001 in. in 10 inches.)

Screw Threads.

U. S. Standard.						Whitworth.	
		Tap drills.		Nuts—rough.			
Diam. in in., d.	Threads per in., n.	U. S.	"V."	Hex.—short diam.	Square—long diam.	Threads per in., n.	Diam. at thd. bottom.
1/4	20	3/16	11/64	1/2	0.7	20	0.186
5/16	18	1/4	15/64	19/32	10/12	18	.241
3/8	16	9/32	9/32	11/16	63/64	16	.295
7/16	14	11/32	21/64	25/32	1 7/64	14	.346
1/2	13	13/32	25/64	7/8	1 15/64	12	.393
9/16	12	7/16	7/16	31/32	1 23/64	12	.456
5/8	11	1/2	1/2	1 1/16	1 1/2	11	.508
3/4	10	5/8	19/32	1 1/4	1 49/64	10	.622
7/8	9	23/32	45/64	1 7/16	2 1/32	9	.733
1	8	27/32	13/16	1 5/8	2 19/64	8	.840
1 1/8	7	15/16	[illegible]	1 13/16	2 9/16	7	.942
1 1/4	7	1 1/16	1 1/32	2	2 53/64	7	1.067
1 3/8	6	1 5/32	1 1/8	2 3/16	3 3/32	6	1.161
1 1/2	6	1 9/32	1 15/64	2 3/8	3 23/64	6	1.286
1 5/8	5 1/2	1 3/8	1 5/16	2 9/16	3 5/8	5	1.369
1 3/4	5	1 1/2	1 13/32	2 3/4	3 57/64	5	1.494
1 7/8	5	1 5/8	1 17/32	2 15/16	4 5/32	4 1/2	1.590
2	4 1/2	1 23/32	1 21/32	3 1/8	4 27/64	4 1/2	1.715
2 1/4	4 1/2			3 1/2	4 61/64	4	1.930
2 1/2	4			3 7/8	5 31/64	4	2.180
2 3/4	4			4 1/4	6	3 1/2	2.384
3	3 1/2			4 5/8	6 17/32	3 1/2	2.634
3 1/4	3 1/2			5	7 1/16		
3 1/2	3 1/4			5 3/8	7 39/64		
3 3/4	3			5 3/4	8 1/8		
4	3			6 1/8	8 41/64		
4 1/4	2 7/8			6 1/2	9 3/16		
4 1/2	2 3/4			6 7/8	9 3/4		
4 3/4	2 5/8			7 1/4	10 1/4		
5	2 1/2			7 5/8	10 49/64		
5 1/4	2 1/2			8	11 23/64		
5 1/2	2 3/8			8 3/8	11 7/8		
5 3/4	2 3/8			8 3/4	12 3/8		
6	2 1/4			9 1/8	12 15/16		

Angle of U. S. and V thread = 60°; Whitworth = 55°, rounded top and bottom. Diam. at bottom of U. S. thread = $d-(1.299 \div n)$. Diam. at bottom of V thread = $d-(1.733 \div n)$. In U. S. threads $\frac{1}{8}$ of height is flatted off at top of thread and bottom is filled in $\frac{1}{8}$ ht. ∴ wearing surface = 0.75 × pitch = 0.866 × ht.

Thickness of rough nuts = d. Finished dimensions of hexagon nuts (short diam. and thickness) are $\frac{1}{16}$ in. less than rough dimensions. Radius for rounding Whitworth threads = pitch of threads × 0.1373.

Stubs' Steel Wire Gauge (continued from table on page 121).

No.	Diam.	No.	Diam.	No.	Diam.	No.	Diam.	No.	Diam.
41	0.095	52	0.063	72	0.024	F	0.257	P	0.323
42	.092	54	.055	74	.022	G	.261	Q	.332
43	.088	56	.045	76	.018	H	.266	R	.339
44	.085	58	.041	78	.015	I	.272	S	.348
45	.081	60	.039	80	.013	J	.277	T	.358
46	.079	62	.037	A	.234	K	.281	U	,368
47	.077	64	.035	B	.238	L	.290	V	.377
48	.075	66	.032	C	.242	M	.295	W	.386
49	.072	68	.03	D	.246	N	.302	X	.397
50	.069	70	.027	E	.25	O	.316	Y	.404
								Z	,413

The U. S. Standard and Imperial gauges are respectively the legal standards in the U. S. and Great Britain. Stubs' steel wire gauge is used in measuring steel wire and drill rods.

Wire and Sheet-Metal Gauges.

No.	American, B. & S.	Birmingham—Stubs (iron).	Stubs (steel).	Washburn & Moen—Roebling	Trenton Iron Co.	U. S. Standard.	Imperial.
0000000				0.49		0.5	0.5
000000				.46		.469	.464
00000				.43	.45	.438	.432
0000	0.460000	0.454		.393	.40	.406	.4
000	.409640	.425		.362	.36	.375	.372
00	.364800	.38		.331	.33	.344	.348
0	.324950	.34		.307	.305	.313	.324
1	.289300	.3	0.227	.283	.285	.281	.3
2	.257630	.284	.219	.263	.265	.266	.276
3	.229420	.259	.212	.244	.245	.25	.252
4	.204310	.238	.207	.225	.225	.234	.232
5	.181940	.22	.204	.207	.205	.219	.212
6	.162020	.203	.201	.192	.190	.203	.192
7	.144280	.18	.199	.177	.175	.188	.176
8	.128490	.165	.197	.162	.160	.172	.16
9	.114430	.148	.194	.148	.145	.156	.144
10	.101890	.134	.191	.135	.130	.141	.128
11	.090742	.12	.188	.12	.1175	.125	.116
12	.080808	.109	.185	.105	.105	.109	.104
13	.071961	.095	.182	.092	.0925	.094	.092
14	.064084	.083	.180	.08	.08	.078	.08
15	.057068	.072	.178	.072	.07	.07	.072
16	.050820	.065	.175	.063	.061	.0625	.064
17	.045257	.058	.172	.054	.0525	.0563	.056
18	.040303	.049	.168	.047	.045	.05	.048
19	.035390	.042	.164	.041	.039	.0438	.04
20	.031961	.035	.161	.035	.034	.0375	.036
21	.028462	.032	.157	.032	.03	.0344	.032
22	.025347	.028	.155	.028	.027	.0313	.028
23	.022571	.025	.153	.025	.024	.0281	.024
24	.020100	.022	.151	.023	.0215	.025	.022
25	.017900	.020	.148	.02	.019	.0219	.02
26	.015940	.018	.146	.018	.018	.0188	.018
27	.014195	.016	.143	.017	.017	.0172	.016
28	.012641	.014	.139	.016	.016	.0156	.014
29	.011257	.013	.134	.015	.015	.0141	.013
30	.010025	.012	.127	.014	.014	.0125	.012
31	.008928	.010	.120	.0135	.013	.0109	.011
32	.007950	.009	.115	.013	.012	.0101	.0108
33	.007080	.008	.112	.011	.011	.0094	.01
34	.006304	.007	.110	.010	.01	.0086	.009
35	.005614	.005	.108	.0095	.009	.0078	.008
36	.005000	.004	.106	.009		.007	.007
37	.004453		.103	.0085		.0066	.0068
38	.003965		.101	.008		.0063	.006
39	.003531		.099	.0075			.005
40	.003145		.097	.007			.0048

Imperial Wire Gauge (continued from table).

No.	41	42	43	44	45	46	47	48	49	50
Diam. . . .	.0044	.004	.0036	.0032	.0028	.0024	.002	.0016	.0012	.001

Grinding Wheels. The abrasives used in grinding wheels are corundum, emery (impure corundum), carborundum and alundum. The first two

occur in a natural state, while the latter are products of the electric furnace, are very much harder and have greater cutting power and durability. Carborundum (SiC) is composed of 30% Carbon+70% Silicon. Alundum is obtained principally from bauxite, an amorphous hydrate of alumina.

Speeds. Peripheral speeds of wheels vary from 3,000 to 7,000 ft. per min., usually from 5,000 to 6,000. Cylindrical work in grinding-machines should have a peripheral speed of from 25 to 80 ft. per min., the slower speeds for delicate work. The traverse speed of wheel = face of wheel × 0.75 per rev. of piece being ground. Polishing wheels should have a peripheral speed of about 7,000 ft. per min.

Grades of Wheels for Various Uses. Abrasives are classified (according to the size of their grains) by numbers which indicate the meshes per linear inch of the screen through which the crushed substance has passed.

The cutting capacity of the various sizes compared with files is as follows: 16 to 30, rough files; 30 to 40, bastard; 46 to 60, second-cut; 70 to 80, smooth-cut; 90 and upwards, superfine to dead-smooth.

The Norton Emery Wheel Co. gives the following table which is approximately correct for ordinary conditions. (I = medium soft wheel, M = medium, Q = medium hard; other letters indicate corresponding intermediate grades):

	No. of grain.
Large C. I. and steel castings (Q, R)	12 to 20
Large malleable and chilled iron castings (Q, R)	16 to 20
Small castings (C. I., steel and malleable iron), drop-forgings (P, Q)	20 to 30
W. I., bronze castings, plow points (P, Q), brass castings (O, P)	16 to 30
Planer and paper-cutter knives (J, K), lathe and planer tools (N, O)	30 to 40
General machine work (O, P)	30 to 40
Wood-working tools, saws, twist-drills, hand-ground (M, N)	36 to 60
Machine grinding: twist drills (K, M), reamers, taps, milling cutters (H, K)	40 to 60
Hand grinding: reamers, taps, milling cutters (N, P)	46 to 100

For grinding machines, the Landis Tool Co. gives the following:

Material.	No. of grain.	Grade of wheel.
Soft steel, ordinary shafts	24 to 60	Medium or one grade harder.
" " tubing or light shafts	24 " 60	One or two grades softer than medium.
Hard steel and C. I.	24 " 60	Medium or one grade softer.
Internal grinding	30 " 60	" to several grades softer.

Economy in Finishing Cylindrical Work is obtained by reducing stock by means of rough, heavy cuts to within .01 to .025 in. of the finished diameter and then grinding to completion. It is possible to force wheels to remove 1 cu. in. per min.

Emery Wheels vs. Filing and Chipping. The figures in the following table express approximately the number of lbs. removed per hour by the various processes. The metal bars ground were ¾ in. × ½ in., held against wheel by a pressure of about 100 lbs. per sq. in. (T. Dunkin Paret, Jour. Franklin Inst., 5-12-1904):

	Brass.	C. I.	W. I.	Hardened Saw Steel.
Emery wheel	34.	15.5	5.	6.87
File	1.	.72	.34	.125
Cold chisel	2.56	4.69	1.31	.187
Wheel wear	.8	1.37	1.69	3.63

Grindstones for tool-dressing should have a peripheral speed between 600 and 900 ft. per min. Rapid grinding speeds should not exceed 2,800 ft. per min.

High-Speed Tool Steel. In 1900 the Bethlehem Steel Co. exhibited tool steel at the Paris Exposition made and treated according to the Taylor-White patents. This steel was capable of taking heavy cuts at abnormally high cutting speeds, the chips coming off at a red heat, and the tool stand-

ing up well under the work. Since that date many steels of similar capacity have been placed on the market by various makers.

These steels are air-hardening and contain (in addition to carbon) one or more of the elements, chromium, tungsten, vanadium, molybdenum, and manganese, these elements uniting with the carbon to form carbides. Iron carbides exist generally in an unhardened state and at high temperatures these part with their carbon, which then shows a greater affinity for chromium, etc. These newly formed carbides may be fixed by rapid cooling, and they impart the extraordinary hardness which they possess to the steel. This hardness is retained by the steel, as these carbides are not affected by changes of temperature within certain limits. Tools made from these steels are forged at a bright red heat and slowly cooled. The points are then reheated to a white, melting heat (about 2,000° F.), cooled to a red heat in an air-blast, and then slowly cooled, or quenched in oil.

Cutting Speeds for High-Speed Tool Steels. Experiments have been conducted in Germany and also in England (by Dr. Nicholson of Manchester) to determine the best cutting speeds to employ on various metals, and the results are expressed by the following formula: Cutting speed in feet per minute, $S=\frac{K}{a+L}+M$, where a is the sectional area of cut in sq. in. (= depth × traverse in one rev.), and K, L, M are constants:

	Whitworth Fluid (Manchester) Pressed Steel			Cast Iron.			W. I.
	Soft.	Medium.	Hard.	Soft.	Medium.	Hard.	
K =	1.95	1.85	1.03	3.1	1.65	1.3	2.62
L =	0.011	0.016	0.16	0.025	0.03	0.035	0.0092
M =	15	6	4	8	7	5.5	23.5

	Siemens-Martin Steel (Berlin).			Cast Iron.	Cast Steel.
	Soft.	Medium.	Hard.		
K =	4.03	0.918	1.17	0.196	0.2
L =	0.012	0.009	0.0075	−0.0199	−0.005
M =	−26	16	−20	32.2	11.25

The chemical composition of the metals experimented upon is as follows:

CAST IRON.

	Berlin.	Manchester.		
		Soft.	Medium.	Hard.
Carbon, combined........	0.45	0.459	0.585	1.15
Graphite................	3.46	2.603	2.72	1.875
Si......................	2.05	3.01	1.703	1.789
Mn......................	1	1.18	0.588	0.348
S.......................	0.1	0.031	0.061	0.1614
P.......................	0.1	0.773	0.526	0.732
Crushing strength in tons of 2,240 lbs.		26.9	44	43.5

STEEL.

	Siemens-Martin.			Whitworth.		
	Soft.	Medium.	Hard.	Soft.	Medium.	Hard.
Carbon..........	0.3	0.54	0.63	0.198	0.275	0.514
Si..............	0.05	.21	.20	.055	.086	.111
Mn..............	.58	.93	1.22	.605	.65	.792
S...............	.05	.025	.05	.026	.037	.033
P...............	.07	.05	.05	.035	.043	.037
Tensile strength in tons (2,240 lbs.).	26 to 32	40	49	26	29	47

Turning. The following results have been taken from the exhaustive presidential address delivered before the A. S. M. E., December, 1906,

and embody the practical conclusions of an investigation by Mr. F. W. Taylor, extending over some twelve years.

Tools Used: round nose. For blunt tools, radius of point $= \frac{1}{2}$ width $- \frac{5}{32}$ in.; for sharp tools, radius $= \frac{1}{2}$ width $- \frac{3}{16}$ in. Clearance angle $= 6°$; back slope $= 8°$; side slope $= 14°$ for hard steel and C. I. ($= 22°$ for medium and soft steel).

Depth of Cut in Ins.	Feed in Ins.	Cutting Speed in Feet per Minute for a Tool which is to last 90 Minutes before Regrinding.							
		Soft Cast Iron.				Soft Steel.			
		Sizes of Standard Tools.				Sizes of Standard Tools.			
		$1\frac{1}{4}$ in.	1 in.	$\frac{3}{4}$ in.	$\frac{1}{2}$ in.	$1\frac{1}{4}$ in.	1 in.	$\frac{3}{4}$ in.	$\frac{1}{2}$ in.
$\frac{3}{32}$	$\frac{1}{64}$	239	226	222	206	518	490	482	510
	$\frac{1}{32}$	191	177	169	147	366	339	323	322
	$\frac{1}{16}$	142	130	120	97.5	257	235	217	203
	$\frac{3}{32}$	118	107	97.0	76.0	209	189	172	
	$\frac{1}{8}$	103	92.8	83.4	64.1				
	$\frac{3}{16}$	85.0	75.7	66.4					
$\frac{1}{8}$	$\frac{1}{64}$	216	205	203	194	450	427	423	445
	$\frac{1}{32}$	172	160	156	138	317	296	284	281
	$\frac{1}{16}$	128	118	110	93.1	223	205	190	177
	$\frac{3}{32}$	107	97.0	88.8	72.1	182	165	151	135
	$\frac{1}{8}$	93.4	84.2	76.2	41.8	157	142	128	
	$\frac{3}{16}$	76.8	68.6	60.9					
$\frac{3}{16}$	$\frac{1}{64}$	187	181	181	182	370	358	358	404
	$\frac{1}{32}$	149	142	137	128	260	247	240	255
	$\frac{1}{16}$	111	104	97.7	86.1	183	171	161	161
	$\frac{3}{32}$	92.5	85.8	78.0	67.4	149	138	127	
	$\frac{1}{8}$	73.1	74.3	67.5		129	118		
	$\frac{3}{16}$	66.4	60.6	54.2		105	95.0		
$\frac{1}{4}$	$\frac{1}{64}$	168	165	167	173	322	315	320	359
	$\frac{1}{32}$	134	129	126	122	227	218	215	226
	$\frac{1}{16}$	99.8	94.3	90.8	81.9	159	150	144	
	$\frac{3}{32}$	83.2	77.8	72.7		130	121		
	$\frac{1}{8}$	72.6	67.5	62.7		112	104		
	$\frac{3}{16}$	59.7	55.0			91.4			
$\frac{3}{8}$	$\frac{1}{64}$	144	143	150		264	263	276	330
	$\frac{1}{32}$	115	112	113		186	182	185	
	$\frac{1}{16}$	85.1	81.9	81.0		131	126		
	$\frac{3}{32}$	70.9	67.6	65.5		107	101		
	$\frac{1}{8}$	62.0	58.6			92.2			
	$\frac{3}{16}$	51.0	57.5						
$\frac{1}{2}$	$\frac{1}{64}$	131	132			230	232		
	$\frac{1}{32}$	105	104			162	161		
	$\frac{1}{16}$	77.6	75.8			114	111		
	$\frac{3}{32}$	64.7	62.6			92.6			
	$\frac{1}{8}$	56.6	54.2						
	$\frac{3}{16}$	46.5	44.2						
$\frac{3}{4}$	$\frac{1}{64}$	112							
	$\frac{1}{32}$	89.2							
	$\frac{1}{16}$	66.2							
	$\frac{3}{32}$	55.2							
	$\frac{1}{8}$	48.3							
	$\frac{3}{16}$	39.7							

Speed for
Hard Cast Iron $= 0.29 \times$ Speed for Soft C. I.
Medium " " $= 0.5 \times$ " " " "
Hard Steel $= 0.23 \times$ " " " steel
Medium " $= 0.5 \times$ " " " "

(Condensed from Tables 143–154, Vol. 28, Proc. A. S. M. E.)

Average composition of tool-steel: 0.3% V + 18% W + 5.76% Cr + 0.678% C + 0.09% Mn + 0.046% Si. Forged at light yellow heat.

Steels Turned. Hard steel (locomotive tire): 0.64% C + 0.7% Mn + 0.21% Si + 0.044% P. Tensile strength = 118,500 lbs. per sq. in. Elastic limit = 70,000 lbs. per sq. in. Per cent. stretch = 14. Medium steel: 0.34% C + 0.6% Mn + 0.183% Si + 0.032% S + 0.035% P. T. S. = 72,830; E. L. = 34,630; stretch = 30%; contraction = 48.7%. Soft steel: 0.22% C + 0.42 Mn + 0.07% Si + 0.025% S + 0.022 P. T. S. = 56,250; E. L. = 26,500; stretch = 35.5%; contraction = 56.3%.

Pressures on Cutting Tools, p, in lbs. per sq. in.

Cast Iron: soft, 115 000; medium, 188,000; hard, 184,000.
Steel: soft, 258,000; medium, 242,000; hard, 336,000.

Metal Removed in Unit Time.

Cast Iron: lbs. per min. = 3.13 Sa; lbs. per hour = 187.8 Sa.
Steel: lbs. per min. = 3.4 Sa; lbs. per hour = 204 Sa.

Power Required by Cutting Tools (lathes, planers, shapers, boring mills). H.P. = $paS \div 33{,}000$. For milling machines J. J. Flather states that H.P. = cw, where w = lbs. removed per hour, and c = 0.1 for bronze, 0.14 for C. I. and 0.3 for steel.

Best Tool Angles. Dr. Nicholson indicates in his dynamometric experiments that the tool edge (in plan) should be at an angle of 45° to the center line of the work, the clearance from 5 to 6°, the tool angle about 65° for medium steel (75° for C.I.) and the top-rake 20° for medium steel (9° for C.I.). (A. S. M. E., Chicago, 1904.)

Average cutting stress: C.I., 150,000 lbs. per sq. in.; steel, 180,000 lbs. H.P. = cutting stress $\times a \times S \div 33{,}000$.

Cutting H.P. for 1 lb. per min. = 1.46 for C.I. and 1.6 for steel.

H.P. lost in tool friction = 0.3 H.P. per lb. per min. ∴ Gross H.P. = 1.76 for C.I. and 1.9 for steel.

The surfacing force for best shop angle (70° for steel) = 67,000 lbs. per sq. in. of cut; similarly, traversing force = 20,000 lbs. per sq. in. The surfacing force will thrust the saddle against the bed if the coefficient of friction equals or exceeds 0.333. The total net force to be overcome by the driving mechanism of the carriage for cutting steel = $(67{,}000 \times 0.333) + 20{,}000 = 42{,}333$ lbs. per sq. in. of cut. Round-nose tools are preferably used.

High-Speed Twist Drills. Power required ∝ r.p.m.; thrust ∝ feed per rev. Thrust increases more rapidly than the power consumed, consequently less power is required to drill a given hole in a given time by increasing the feed than by increasing the r.p.m. The angle of drill-point may be decreased to as low as 90° (standard angle = 118°), thereby reducing the thrust 25% and without affecting the durability of point. (W. W. Bird & H. P. Fairfield, A.S.M.E., Dec., 1904.)

Metal-Cutting Circular Saws. Cutting cold metal: diam., 32 in.; thickness, 0.32 in.; width of teeth (cutting edge), 0.44 in.; teeth 0.2 to 0.5 in. apart; circumferential velocity, 44 ft. per min.; feed, 0.005 to 0.01 in. per sec.

Cutting metal at red heat: diam., 32 to 40 in.; thickness, 0.12 to 0.16 in.; teeth 0.8 to 1.6 in. apart; depth of teeth, 0.4 to 0.8 in.; circumf. vel., 12,000 to 20,000 ft. per min. (Ing. Taschenbuch).

Taylor-Newbold Saw, with inserted teeth of high-speed steel: A $9\frac{1}{4}$ in. cold saw at 76 r.p.m. will cut through $1\frac{3}{4}$ in. hex. cold-rolled steel in 26 seconds, and at 96 r.p.m., in 22 secs. A 36 in. saw, $\frac{7}{16}$ in. thick, teeth averaging $\frac{9}{16}$ in. thick, running at a cutting speed of 85 ft. per min. will cut off a bar of 0.35 carbon steel 14 in. $\times 8\frac{3}{4}$ in. in 20 min. A bar of 0.40 carbon steel $5 \times 5\frac{1}{4}$ can be cut in 4.4 min.

Fits (Running, Force, Driving, Shrink, etc.). In the following table, which is derived from good practice, the first column gives the nominal diameter of hole. The mean value for each class of fit is given and also the permissible variation above or below same. For force, shrink, and driving fits the values given are those by which the diameter of the piece should exceed that of the hole, while for running and push fits they are the values by which the diameter of the hole should exceed that of the piece. Force and shrink fits are given the same value. Push fits are those in which the piece is forced to place by hand-pressure. Running fits are given three values: A, for easy fits on heavy machinery; B, for average high-speed shop practice; C, for fine tool work.

Diam. in in.	Force +		Drive +		Push −		Running −					
							A.		B.		C.	
	Mean	Var.	Mean	Var.	Mean	Var.	Mean	Var.	Mean	Var.	Mean	Var.
0.5	.75	.25	.37	.12	.5	.12	1.5	.5	1	.25	.6	.12
1	1.75	.25	.87	.12	.75	.25	2	.75	1.5	.5	1	.25
2	3.5	.5	1.25	.25	1.25	.25	2.6	.87	1.9	.62	1.15	.4
3	5.25	.75	2	.5	1.75	.25	3.1	1.1	2.3	.8	1.5	.5
4	7	1	2.5	5	1.75	.25	3.8	1.2	2.7	.85	1.6	.6
5	9	1	3	.5	2.25	.25	4.4	1.4	3.1	.9	1.87	.62
6	11	1	3.5	.5	2.25	.25	5	1.5	3.5	1	2	.75

The values above given are in thousandths of an inch; thus, for a driving fit in a hole of 4 in. diam., the piece should be 4.0025 in. in diam. (It may be either 4.002 in. or 4.003 in. and still be within the permissible variation of 0.0005 in. either way.) For locomotive tires and other large shrunk work, Allowance in thousandths of an inch = ($\frac{17}{16}$ × diam. in in.) + 0.5. (S. H. Moore, A.S.M.E., 1903.)

Sizes above 6 in. Diam.: For shrink fits add 0.0025 in. to diam. of piece for each inch of diam. of hole where the part containing the hole is thick and unyielding. Where the metal around the hole is thin and elastic, add 0.0035 in. per in. of diam. For force fits multiply diam. of hole by 1.0007 and add 0.004 in.; variation of 0.001 in. is permissible. For drive fits allow one-half of the excess just given for force fits. For running fits, multiply diam. of hole by 0.000125, add 0.00225 in. and subtract this sum from diam. of hole, thus giving diam. of piece. Variation of 0.001 in. permissible.

Power Required by Machinery.

Machine.	Material.	No. of tools.	H.P. working.	H.P. light.
Wheel lathe, 84 in.	C. I.	2	6	1.5
Boring mills, 54 to 78 in. . .	C. I.	1	4.5 to 6.5	2.5
Slotting machines, 36×12 and 40×15.	W. I.	1	5.3 & 7.3	1.5 & 2.2
Planers:				
Sellers, 62 in.×35 ft. . . .	W. I.	2	24.5	5.8
" 36 in.×12 ft. . . .	"	2	12.5	3
" 56 in.×24 ft.	"	2	16.8	6
Radial drill, 42 in.	"	2 in. drill	2.1	1.1
Shaper, 19 in. stroke	"	1	7.3	1.8

(Baldwin Loco. Works; measurements by separate electric motors.)

Machine.	H.P. of motor required to operate under best conditions.
Niles planer, 10 ft.×10 ft.×20 ft.	30
Pond " 8 " × 8 " ×20 "	25
" " 5½ " × 5 " ×12 "	15
Gray " 28 in.×32 in ×6 ft.	3
Gisholt turret lathe, 28 in. swing.	4
W. F. and J. Barnes drill press, 21 in . . .	1
Niles radial drill, 60 in. arm.	2
Emery Grinder, two 18-in. wheels in use, 950 r.p.m.	5
Pond Vertical Boring Mill, 10-ft. table.	12
Bement & Miles Slotter.	7
Jones & Lamson Turret Lathe, 2 in.×24 in.	1.5
Gisholt Tool Grinder.	4
Hendey-Norton Lathe, 16 in.	2
Putnam Lathe, 18 in.	2.1
Pond " 36 in.	10

(F. B. Duncan, Engineers' Society of W. Pa.)

		Motor H.P.
Punch and Shears,	$1\frac{1}{4}$-in. hole in 1-in. plate, 6 H.P. } shearing 1-in. plate, 15 H.P. }	12
Plate-edge Planer,	35 ft. × 1 in.	30
	15 ft. × $\frac{3}{4}$ in.	25
Wood Planers	....	4–16
Circular Saws	....	4–24

(D. Selby Bigge.)

H. P. of Motors for Machine Tools. Ordinary lathes: H.P. $= 0.15S - 1$; Heavy lathes and boring mills under 30 in.: H.P. $= 0.234S - 2$; Boring mills over 30 in. swing: H.P. $= 0.25S - 4$; Ordinary drill presses: H.P. $= 0.06S$; Heavy radial drills: H.P. $= 0.1S$; Milling machines: H.P. $= 0.3W$; Planers (2 tools), ordinary: H.P. $= 0.25W$; Do., heavy; H.P. $= 0.41W$ (Ratio of planer feed to return $= 1:3$). Slotters: 10 in. stroke, H.P. $= 5$; 30 in. stroke, H.P. $= 10$; Shapers: 16-in. stroke, H.P. $= 3$; 30 in. stroke, H.P. $= 6.5$.

In the above S = swing in inches and W = width between housings in inches. Formulas based on the cutting by ordinary water-hardened steel tools at 20 ft. per min. (J. M. Barr, in Electric Club Journal.)

If high-speed steels are used, the power required will be from 2.5 to 3 times the above figures on account of increased speeds and cuts.

Power Absorbed by Shafting. In cotton and print mills about 25% of the total transmission; in shops using heavy machinery, from 40 to 60%. In average machine-shops 1 H.P. is required for every three men employed.

COST OF POWER AND POWER PLANTS.

Water Power. Cost of plant per H.P., including dam, $60.00 to $100.00; without dam, $40.00 to $60.00. Power costs from $10.00 to $15.00 per year per H.P.

Steam Power. Cost of engines per H.P.: Simple, slide-valve, $7.00 to $10.00; simple Corliss, $11.00 to $13.00; compound, slide-valve, $12.00 to $15.00; compound Corliss, $18.00 to $23.00; high-speed automatic, $10.00 to $13.00; low-speed automatic, $15.00 to $17.00. Plain tubular boilers, per H.P., $10.00 to $12.00; water-tube boilers per H.P., $15.00. Pumps, $2.00 per H.P. for non-condensing, and $4.00 for condensing. (Dr. Louis Bell in "The Electrical Transmission of Power.") Total cost of plant ranges from $50.00 to $75.00 per H.P., exclusive of buildings.

Dynamos and other electrical apparatus, including switch-boards, cost from $20.00 to $35.00 per kilowatt capacity ($15.00 to $26.00 per H.P.), making the cost of an electrical power plant range from $65.00 to $100.00 per H.P.

The cost of a H.P. hour has been estimated by various authorities to range from 0.55 to 0.85 cents. Dr. Bell places it at 0.8 to 1.00 cent with large, compound-condensing engines, and at 1.5 to 2.5 cents with simple engines, basing his calculations on a day of 10 hours, under full load. If the load is fractional and irregular, these figures should be altered to 1.00 to 1.5 cents and to 3 and 4 cents, respectively.

The cost of electric power includes the cost of steam power to operate the generators, interest, repairs and depreciation on the apparatus, attendance, etc. In very large power plants under good load conditions the cost per kilowatt hour (1.34 H.P. hour) may be as low as one cent, at the bus bars.

Gas Power. The cost of plant is about the same as that of a steam plant. The gas consumption per brake H.P. per hour is about as follows: Natural gas, 10 to 12 cu. ft.; coal gas, 16 to 22 cu. ft.; producer gas, 90 cu. ft.; blast-furnace gas, 116 cu. ft. Coal consumption when producer gas is used is about 1.25 lbs. per B.H.P. With dollar gas, 1 B.H.P. costs 2 cts. per hour. One B.H.P. in a gasoline engine costs about 1.5 cents per hour, in an oil-engine about 1.75 cts. per hour, and in a Diesel engine from 1 to 2 cents, according to the cost of oil in the locality.

Proportions of Parts in a Series of Machines. When two sizes of a machine have been constructed and it is desired to extend the series or to introduce intermediate sizes, the following method of Dr. Coleman Sellers may be employed:

Let D be the larger nominal dimension, say 30 (of a 30-in. swing lathe)
" D_1 " " smaller " " " 12 (" 12-in. " ")

Let diam. of lead-screw on $D=C=3$ in., and diam. of lead-screw on $D_1=C_1=1.5$ in. Thn $D-D_1=30-12=18$, and $C-C_1=3-1.5.=1.5$. $(C-C_1)\div(D-D_1)=1.5\div 18=0.0833=A$, a factor. $AD_1=0.0833\times 12=1$. $C_1-AD_1=1.5-1=0.5=I$, the increment.

Let it be desired to find C_2 when $D_2=20$ in. Then

$$C_2=(D_2\times A)+I=(20\times 0.0833)+0.5=2.16 \text{ in.}$$

Hoisting Engines. Theoretical H.P. required = weight in lbs. (of cage, rope, and load) × speed in ft. per min. ÷ 33,000. Add 25 to 50% for actual H.P. on account of friction and contingencies. Max. limit of rope length in ft. $x=\frac{f}{7w}-\frac{D}{w}$, where f is the breaking strength of rope in lbs. per sq. in., w = lbs. per foot of rope, D = dead weight to be lifted, in lbs., and 7 = factor of safety.

Elevators. Speeds: low, 0 to 150 ft. per min.; medium, 150 to 350 ft.; high, 350 to 800 ft. Counterweights should be about 75% of the weight of car and plunger. Floor area, 20 to 40 sq. ft. Number of elevators for a high office building = (Height of building in ft. × 330) ÷ (speed in ft. per min. × interval between elevators in seconds). (G. W. Nistle, A. S. M. E., May, '04.)

Wire ropes for elevators (6 strands, each of 19 wires): Safe working load in lbs. $=11{,}600d^2-720{,}000\frac{d^3}{D}$ (for Swedish iron); $=23{,}200d^2-760{,}000\frac{d^3}{D}$ (for cast steel), where d = diam. of rope in in. and D = diam. of sheave in in. (Capt. H. C. Newcomer, U. S. A., E. N., 1–15–03.)

Conveyor Belts. Lbs. conveyed per min. $=b^2wV\div 13{,}824$; lbs. per hour $=b^2wV\div 230.4$; tons per hour $=b^2wV\div 460{,}800$, where b = width of belt in in., V = speed in ft. per min., w = lbs. in 1 cu. ft. of the substance conveyed. These values are for flat belts; for trough belts multiply by 3. Average $V=300$; higher speeds may be used, up to 450 for level and 650 when elevating at an angle. Approx. H.P. required to operate = lbs. per min. × elevation in ft. ÷ 16,500.

Electric Cranes. An electric travelling crane consists of a bridge, or girder, a trolley running on the bridge and a hoist attached to the trolley, each part being operated by its own motor. The following data are from a paper by S. S. Wales, read before the Engineers' Society of W. Pa.

L = working load on crane, in tons; W = weight of bridge, in tons; w = weight of trolley, in tons; S = speed in feet per min.; P and P_1 = tractive force in lbs. per ton.

Bridge.			Trolley.		
Span.	W.	P.	L.	w.	P_1.
25 ft.	$0.3L$	30 lbs.	1 to 25 tons	$0.3L$	30 lbs.
50 "	$.6L$	35 "	25 " 75 "	$.4L$	35 "
75 "	$1.0L$	40 "	75 " 150 "	$.5L$	40 "
100 "	$1.5L$	45 "			

H.P for bridge $=PS(L+W+w)\div 33{,}000$. (Use motor 1.5 times as large.)
H.P for trolley $=P_1S(L+w)\div 33{,}000$ (" " 1.25 " " ")
H.P. for hoist $=LS\div 10$ (= 1 H.P. per ton lifted 10 ft. in one minute).

Speeds in Feet per Minute (Ing. Taschenbuch).

	5 tons.	25 tons.	50 tons.	100 tons.
Hoist.	14 to 28	10 to 12	6 to 7.5	5
Bridge.	180 " 300	140 " 210	130 " 200	120
Trolley.	80 " 120	50 " 75	35 " 55	25 to 35

Paint and Painting.

One gallon of linseed oil plus 40 lbs. of white lead will cover 250 to 350 sq. ft. of outside work with a good first coat. The same quantity will second-coat and finish from 350 to 450 sq. ft. White lead when used on inside work turns blackish-yellow on account of exposure to the sulphurous fumes from gas or coal. White zinc is accordingly preferable for inside work, but, having less opacity, more coats are required.

For iron- and steel-work red lead (40 lbs. per gal. of oil) is an expensive but durable covering. To prevent blistering on outside work boiled oil should be used. Turpentine only should be used for thinning. Knots and pitchy surfaces on wood should be coated with shellac varnish, and all grease, scale, acid, and moisture should be removed from metal work before painting. Graphite mixed with linseed oil and laid on in fairly thick coats makes a good paint for metals. Iron pipes, stacks, boiler fronts, etc., are varnished with asphaltum thinned with turpentine.

ELECTROTECHNICS.

ELECTRIC CURRENTS.

Resistance (symbol R) is that property of a material which opposes the flow of an electric current through it. The unit of measurement is the ohm, which is a resistance equal to that of a column of pure mercury at 0° C., of uniform cross-section, 106.3 centimeters in length and weighing 14.4521 grams.

Electro-motive Force (symbol E, abbreviation E.M.F.) is the electric pressure which forces a current through a resistance. The unit of measurement is the volt, the value of which is derived from the standard Clark cell whose E.M.F. at 15° C. is 1.434 volts.

Current (I). An E.M.F. applied to a resistance will cause a flow of electricity which is termed a current. The unit of measurement is the ampere, or the current which flows through a resistance of one ohm when it is subjected to an E.M.F. of one volt. One ampere is the amount of current required to electrolytically deposit 0.001118 gram of silver in one second.

Quantity (Q). The quantity of electricity passing through a given cross-section of conductor is measured in coulombs. One coulomb is the quantity of electricity which flows past a given cross-section of a conductor in one second, there being a current of one ampere in the conductor.

Capacity (C) is that property of a material by virtue of which it is able to receive and store up (as a condenser) a certain charge of electricity. A condenser of unit capacity is one that will be charged to a potential of one volt by a quantity of one coulomb. The unit of capacity is the farad, which is too large for convenient use,—the microfarad (one millionth of one farad) being employed in practice.

Electric Energy (W), or the work performed in a circuit through which a current flows, is measured by a unit called the joule. One joule is equal to the work done by the flow of one ampere through one ohm for one second.

Electric Power (P) is measured in watts. One watt is equal to the work done at the rate of one joule per second. One H.P. = 746 watts. One watt = 0.7373 ft.-lbs. per sec., = 0.0009477 B.T.U. per sec. One kilowatt = 1,000 watts = 1.3405 H.P.

Subdivisions and Multiples of Units are expressed by the use of the following prefixes. One-millionth, micro; one-thousandth, milli; one million, meg-a, one thousand, kilo (e.g., microhms, microfarads, milliamperes, megohms, megavolts, kilowatts, etc.).

Aids to a Conception of Electrical Magnitudes. One ohm = resistance of 1,600 ft. of No. 8 copper wire ($\frac{1}{8}$ in. diam.) approx., = resistance of 400 ft. of No. 14 copper wire ($\frac{1}{16}$ in. diam.) approx. One volt = 90% of the E.M.F. of a Daniell cell (Zn, Cu, and a solution of copper sulphate), 66% of the E.M.F. of a Leclanché cell (carbon-zinc telephone battery), approx.

A 2,000 candle-power (c.-p.) direct current arc lamp has a current of about 10 amperes flowing through it, and an E.M.F between the carbons of about 45 volts; it consequently requires 450 watts of electric power.

An ordinary 16 c.-p. incandescent lamp on a 110-volt circuit requires about 0.5 ampere, its resistance being about 220 ohms and its power consumption about 55 watts.

Ohm's Law. If E is the difference of potential (E.M.F.) in volts between two points in a conductor through which a steady, direct current of I amperes is flowing, and the resistance of the conductor between the two points is R ohms, then $I=\frac{E}{R}$, or $E=IR$.

Divided Circuits. If a current arrives at a point where several paths are open to its flow, it divides itself inversely as the resistances of these paths, or directly as their respective conductances. (The conductance of a circuit is the reciprocal of its resistance, or $\frac{1}{R}$.) $i_1:i_2:i_3=\frac{1}{r_1}:\frac{1}{r_2}:\frac{1}{r_3}$, etc., and $i_1+i_2+i_3=I$.

The total conductance of the branched circuits, $\frac{1}{R}=\frac{1}{r_1}+\frac{1}{r_2}+\frac{1}{r_3}$, etc., and the reciprocal of this value equals the joint resistance of the several paths. For two branches $\frac{1}{R}=\frac{1}{r_1}+\frac{1}{r_2}$, and $R=\frac{r_1r_2}{r_1+r_2}$.

Kirchoff's Laws. 1. The sum of the products of the currents and resistances in all the branches forming a closed circuit equals the sum of all the electrical pressures in the same circuit, or $\Sigma E=\Sigma(IR)$. 2. At every joint in a circuit, $\Sigma I=0$, or the sum of the currents flowing toward the joint equals the sum of the currents flowing away therefrom.

Resistance of Conductors. The resistance R (in ohms) of a conductor of length l (in cms.) and cross-section s (in sq. cms.) is $R=cl\div s$, where c is the specific resistance of the material (the resistance between two opposite faces of a cube 1 cm. long and 1 sq. cm. cross-section).

Specific Resistances at 0° C. are given in the following table. When any higher temperature is taken, add as a correction $b\times$degs. C. above 0°.

	Specific resistance in microhms.	b.		Specific resistance in microhms.	b.
Silver	1.468	0.004	Nickel	12.323	0.00622
Copper	1.561	.00428	Tin	13.048	.0044
Gold	2.197	.00327	Lead	20.38	.00411
Aluminum	2.665	.00435	Mercury	94.07	.00072
Zinc	5.751	.00406	German silver	29.982	.000273
Iron	9.065	.00625	Carbon	4,200 to 40,000	−0.2
Platinum	10.917	.003669			

Dilute Sulphuric Acid.

Per cent wt. of H_2SO_4 in solution	5	15	30	45	60	80
Sp. res. at 18° C. in ohms	4.8	1.9	1.4	1.7	2.7	9.9

(For each deg. C. rise in temp. subtract 1.4% from above values.)

Joule's Law. If a current of I amperes flows through a resistance of R ohms for t seconds, the heat developed, $=I^2Rt$, in joules or watt-seconds, $=0.239I^2Rt$ gram-calories, $=0.0009477I^2Rt$ B.T.U.

The heat developed is equivalent to the energy causing the current flow. Rate of expenditure of energy, in watts, $=EI=I^2R$. Energy in joules or watt-seconds $=EIt=I^2Rt$.

Electrolysis is the separation of a chemical compound into its constituent elements by means of an electric current. Two plates or poles (electrodes) are inserted in the compound or electrolyte, the electrode of higher potential being called the anode, and the other the cathode. The products of the decomposition are called ions. A current I amperes flowing through an electrolytic bath will deposit a weight of G grams in t units of time.

$G=kaIt$, where a is the chemical equivalent of the substance.

If t is in seconds, k=0.000010386; if t is in minutes, k=0.0006232, and if t is in hours, k=0.03739. The electro-chemical equivalent=grams per coulomb.

	a.	Grams per coulomb.	Grams per amp. hour.		a.	Grams per coulomb.	Grams per amp. hour.
Aluminum	9	0.00009347	0.3365	Oxygen...	8	0.00008309	0.2991
Copper....	31.6	.00032820	1.1815	Platinum..	97.2	.00100952	3.6343
Gold......	65.4	.00067924	2.4453	Potassium.	39	.00040505	1.4582
Lead......	103.2	.00107184	3.8585	Silver....	107.7	.00111857	4.0269
Mercury...	99.9	.00103756	3.7352	Tin.......	58.7	.00060966	2.1948
Nickel....	29.3	.00030431	1.0955	Zinc......	32.4	.00033651	1.2114
Nitrogen..	4.6	.00004840	0.1742				

(To obtain pounds per ampere-hour, multiply grams per ampere-hour by 0.0022046.)

ELECTRO-MAGNETISM.

Lines of Force. When a current starts to flow in a conductor, whirls of magnetism are generated around the conductor which seem to spring from its center, and the region so filled with these whirls increases radially in extent as the current increases, remains constant when a steady current is attained, and shrinks radially to nil when the current is interrupted.

If the conductor is bent into a loop, an elementary electro-magnet is formed, with a pole on either side of the plane of the loop. If the conductor be wound into a number of loops along the surface of a cylinder, a solenoid is formed and the whirls so add themselves together that they may be considered as loops, entering the solenoid at all points of the section at one end, passing along inside parallel to the axis of the solenoid to the other end, thence emerging and returning outside in curved paths to the point first considered (Fig. 27).

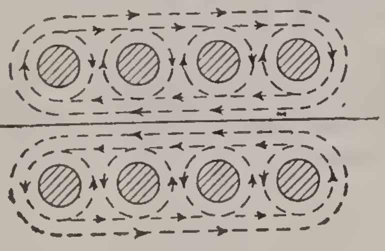


Fig. 27.

These loops are termed lines of force, and their number depends on the number of spirals of conductor in the solenoid and the number of amperes of current flowing through them, or, as it is expressed, by the number of ampere-turns.

The Intensity of the Magnetic Field ($\mathcal{H}$) at any point is measured by the force it exerts on a unit magnetic pole, the unit intensity, therefore, being that which acts with a force of one dyne upon a unit pole, or one line of force per sq. cm. (A dyne is the force which, acting for one second upon a mass of one gram, imparts a velocity of one centimeter per second.)

Magneto-motive Force ($\mathfrak{F}$) is the magnetizing force of an electric current flowing in a coil or solenoid and is usually stated in ampere-turns. $\mathfrak{F} = 4\pi nI \div 10 = 1.257nI$, where n is the number of turns or loops of the conductor and I the current in amperes. The unit for $\mathfrak{F}$ is called the gilbert and is equal to 0.7958 ampere-turns.

The Intensity of the Magnetizing Force per unit length of solenoid $(\mathcal{H}) = 4\pi nI \div L = 1.257nI \div L$, where L=length in cm. If L_1=length in inches, $\mathcal{H} = 0.495nI \div L_1$ or, if expressed in lines per sq. in., $\mathcal{H}_1 = 3.193nI \div L_1$.

* **Magnetic Induction** ($\mathcal{B}$) is the magnetic flux or the number of lines of force per unit area of cross-section, the area at every point being normal to the direction of the flux. $\mathcal{B} = \mu\mathcal{H}$, where μ is the permeability. The unit is the gauss, or one maxwell per normal sq. cm.

The Magnetic Flux (Φ) is equal to the average field intensity×area. The unit is the maxwell, or the flux due to unit magneto-motive force (M.M.F.) when the reluctance is one oersted.

Reluctance ($\mathcal{R}$) is the resistance offered to the magnetic flux by the material undergoing magnetization. The unit is the oersted, or the resistance offered by one cubic centimeter of vacuum.

Magnetic Susceptibility, $(\kappa) = \mathfrak{F} \div \mathcal{H}$.

* B, F, and H are commonly used in place of $\mathcal{B}$, $\mathfrak{F}$, and $\mathcal{H}$.

Reluctivity (ν) is the reluctance per unit of length and unit cross-section, $=1 \div \mu$. Maxwells = gilberts ÷ oersteds.

Hysteresis. When a magnetic substance (e.g., iron) is magnetized, the intensity of magnetization does not increase as rapidly as does the magnetizing force, but lags behind it. This tendency is termed hysteresis, and it may be considered as an internal magnetic friction of the molecules of the substance. Continued rapid magnetizing and demagnetizing will cause the substance to become heated. Hysteresis (h) may be calculated by the following formula due to Steinmetz: h (in watts) $=\eta\mathfrak{B}^{1.6}kn10^{-7}$, where k = volume in cu. cms. and n = number of complete cycles of magnetization and demagnetization per second.

	η.		η.
Very thin, soft sheet iron. . . .	0.0015	Soft, annealed cast steel.	0.008
" soft iron wire.	.002	" machine steel.	.0094
Thin sheet iron (good)	.003	Cast steel.	.012
Thick " "	.0033	Cast iron.	.016
Ordinary sheet iron.	.004	Hardened cast steel.	.025

The Magnetic Circuit. Magnetism may be considered as flowing in a magnetic circuit in the same manner as an electric current does in a conductor and the following relation holds:

Magnetic Flux $=\dfrac{\text{Magneto-motive Force}}{\text{Reluctance}}$, which is analogous to Current $=\dfrac{\text{E.M.F.}}{\text{Resistance}}$.

$\Phi=\mathfrak{F}\div\mathfrak{R}$. Reluctance, $\mathfrak{R}=l\div\mu a$, where l = length of magnetic circuit, a = area of cross-section and μ = permeability (see Dynamos). $\Phi=\mathfrak{F}\div\mathfrak{R}$, $\mathfrak{F}=1.257nI$; $\therefore\ nI=\dfrac{\Phi l}{\mu a\div 1.257}=0.7958\Phi\dfrac{l}{\mu a}$, where l is in cms. and a in sq. cms. When l_1 and a_1 are in inch measure, $nI=0.3132\Phi l_1\div\mu a_1$.

Induction. If a conductor, of length dl, is moved in a magnetic field (of strength $\mathfrak{H}$) with a velocity, v (the conductor making the angle α with the direction of the lines of force and the direction of motion being at the angle β with the plane passing through the conductor in the direction of the lines of force), the induced electromotive force, $dE=\mathfrak{H}v\sin\alpha\sin\beta dl$, or, $E=\int\mathfrak{H}v\sin\alpha\sin\beta dl$. When $\alpha=\beta=90°$, E is a maximum and is equal to $\mathfrak{H}vl10^{-8}$ volts, when v is stated in cms. per sec. and l in cms.

The mean E.M.F. of the armature of a two-pole dynamo, $E=\dfrac{\Phi nN10^{-8}}{60}$ volts, where Φ is the total number of lines of force flowing between the pole-faces, n the number of active conductors on the armature, and N = r.p.m. In a series-wound multipolar dynamo, $E=\dfrac{\Phi_1 pnN10^{-8}}{60}$ volts, and in a multiple-wound multipolar dynamo, $E=\Phi_1 nN10^{-8}\div 60$, where Φ_1 = no. of lines flowing between one pair of poles, and p = no. of pairs of poles.

The Direction of Currents, Lines of Force, etc. The lines of force in a magnet or solenoid flow from the south pole to the north pole and return outside to the south pole. The north pole of a magnetic needle when brought near a magnet points in the direction of the lines of force.

To determine the direction in which a current flows in a conductor, place a compass underneath it. If the north pole of the needle points away from the person holding compass (who is at one side of the conductor) the current is flowing to his right.

To find the direction of a current flowing in a coil, find the north pole by means of a compass, the north pole of which will be repelled by the north pole of the coil or magnet. Then place the right hand on the coil with the thumb (at right angles to the extended fingers) pointing in the direction of the north pole and the current will be flowing in the direction in which the fingers are pointing. If the direction of current is known, the north pole may be similarly determined.

The positive (+) pole of a generator of electric current is the one from which the current flows into the external circuit. In primary batteries the zinc is negative, copper, carbon, etc., being the positive poles.

Direction of an Induced Current.—If the letter N be drawn on the face of a north pole and a conductor (parallel to the vertical lines of the letter) be moved past the pole in a plane parallel to the pole face, the direction of current flow will be determined by the motion of the point of intersection (projected) of the conductor and the oblique line in the letter N. Thus, if the conductor moves from left to right, the point of intersection moves from above to below, which indicates the direction of the induced current.

ELECTRO-MAGNETS.

Traction or Lifting Power. If a bar of iron be bent into the shape of the letter U and coils of insulated wire are wound upon the limbs, the electro-magnet thus formed (when a current is flowing through the coils) will have a lifting or holding power on each limb of P (in lbs.) $=B^2a\div 72{,}134{,}000$, where $B=$no. of lines of force per sq. in. of iron section and a is the area of one pole-face of the magnet. The number of ampere-turns is the coils necessary to produce the pull, $P=nI=2{,}661\,\frac{l}{\mu}\sqrt{P\div a}$, where l is the length of the magnetic circuit in inches and μ the permeability. B may be taken at 110,000 for W. I. and mild steel.

The above formula is used when the keeper or armature is in contact with the pole-faces. If the keeper (by which the weight to be lifted or held is supported) is distant z inches from the pole-faces, then, $nI=2z\times B\times 0.3133$.

If the iron is of good quality and far from saturation the number of ampere-turns required to force the flux through the metal part of the circuit is small enough, comparatively, to be negligible, and the formula value, which is the ampere-turns required to force the flux across the air-gaps, may be taken as the total.

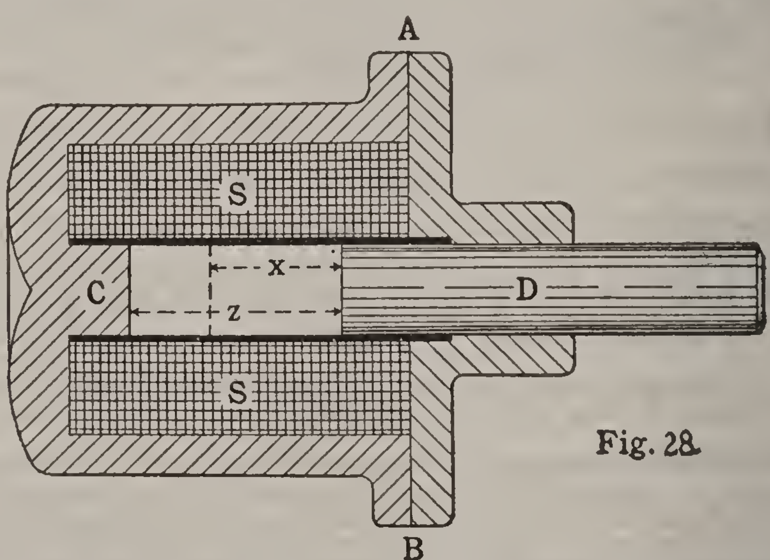


Fig. 28.

An iron-clad magnet which may be similarly considered is shown by the part ABC in Fig. 28; the cylindrical core C, however, should extend through the coil to the plane AB.

Plunger Electro-Magnets. Fig. 28 shows an electro-magnet of the iron-clad or jacketed type, which is provided with a movable plunger or core, D, an inner projecting core, C, and a guide or "stuffing-box," E. The air-gap is indicated by z and x is the stroke of the plunger or its range of motion, which must be less than z in order to meet the conditions imposed in designing for certain specified pulls at the beginning and end of stroke.

Pull in lbs.$=P=aB^2\div 72{,}134{,}000$ (1). $B=nI\div 0.3133z$ (2). Maximum pull (at end of stroke)$=P_g$. Minimum pull (at beginning of stroke)$=P_l$. Let $y=P_g\div P_l=\frac{B_g{}^2}{B_l{}^2}$, then $\sqrt{y}=\frac{B_g}{B_l}$ and $B_l=B_g\div\sqrt{y}$ (3). At

the beginning of stroke, $B_l z \times 0.3133 = nI$, and, at the end of stroke, $0.3133 B_g(z-x) = nI$, consequently

$$\frac{z}{z-x} = B_g \div B_l = \sqrt{y} \quad \text{and} \quad z = x\sqrt{y} \div \sqrt{y} - 1 \ (4).$$

Let d = diam. of core in in., then, $a = 0.7854d^2$, and, from (1), $d = 9{,}580\sqrt{P_l} \div B_l$ (5), which determines d if B_l is fixed upon.

If d is fixed, $B_l = 9{,}580\sqrt{P_l} \div d$ (5a). From (2), $nI = 3{,}000z\sqrt{P_l} \div d$ (6), which allows the calculation of the ampere-turns if d has been decided upon. Length of winding bobbin in in. $= L$; available winding depth in in. $= T$; mean length of one turn in in. $= M$; sectional area of coil in sq. in. $= LT$; winding volume $= MLT$. If the actual permissible current density over the gross section is β, then $nI = \beta LT$, or, $LT = nI \div \beta$ (7). For momentary work β may be from 2,000 to 3,000 amperes, if the magnet is well ventilated and provided with radiating surfaces. For continuous use over several hours, $\beta = 300$ to 400 amp. From (6) and (7), $T = 3{,}000z\sqrt{P_l} \div \beta dL$. Assume that $L = z$, then, if β is taken at 2,000, $T = 1.5\sqrt{P_l} \div d$ (9). $M = \pi(0.25 + d + T)$ (10), assuming that the core of bobbin and clearance add 0.25 in. to d. Current density in copper (amperes per sq. in.) $= \alpha$; diam. of bare wire $= \delta$, do. of insulated wire $= \delta_1$; R = resistance in ohms; r_1 = resistance in ohms per inch of wire; s = sectional area of wire in sq. in.; σ = space factor, = total copper section $\div LT$; V = volts at terminals; w = watts used; $VI = I^2R$. ρ = resistance in ohms per cu. in. of coil space. If I is given, $nI \div I = n$; $\beta = nI \div LT$; $\rho = 0.8\alpha\beta \div (I^2 \times 10^6)$; $s = I \div \alpha$, and $V = w \div I$.

If V is given, $I = w \div V$; $r_1 = V \div MnI$, or, R per 1,000 ft. $= 12{,}000V \div MnI$; $\delta = 0.001\sqrt{MnI \div V}$; $s = 0.8MnI \div V \times 10^6$; $\sigma = 0.7854\delta^2 \div \delta_1^2$; $LT = ns \div \sigma$, and $M = 417dw \div \alpha z\sqrt{P_l}$.

If a solenoid is provided with an ample and well fitting iron guide or stuffing-box at the end at which the plunger enters the coil, the effect of its presence will be to bring up the field at the point when the plunger is just entering to the intensity which exists at mid-length of the solenoid. The maximum pull (when plunger has reached the bottom of the coil) is one-quarter of that calculated from equation (1). If the permeability of the iron is known, B can be found from tables.

Calculation of a Plunger Electro-Magnet. A number of designs should be made and the calculations tabulated in order to determine the most economical one, in weight of copper and in watts required.

Example: It is required to design an iron-clad coil to give an initial pull of 25 lbs., increasing to 100 lbs. at the end of a stroke or range of 2 inches, E.M.F. supplied being 100 volts, for intermittent work.

$P_g = 100$; $P_l = 25$; $x = 2$; $y = 4$; $\sqrt{y} = 2$; $z = 4$; $\sqrt{P_l} = 5$. $nId = 3{,}000 \times 4 \times 5 = 60{,}000$; $B_l d = 9{,}580 \times 5 = 47{,}900$, and $B_g = 47{,}900 \times 2 = 95{,}800$.

	Trial Values.			
d in inches. . . . . . =	1	2	3	4
nI. . . . . =	60,000	30,000	20,000	15,000
B_l. . . . . =	47,900	23,950	15,966	11,975
B_g. . . . . =	95,800	47,900	31,932	23,950

Let $\beta = 2{,}000$, $\sigma = 0.5$; then, $\alpha = 4{,}000$. Then, for $T = < 3$ in. (which will allow from 10,000 to 30,000 amp.-turns per inch length of coil, if properly ventilated)

d in inches. . . . . . =	1	2	3	4
LT. . . . . =	30	15	10	7.5
L. . . . . =	10	5	4	3.75
T. . . . . =	3	3	2.5	2
M. . . . . =	13.36	16.5	18.07	19.65
MLT. . . . . =	400.8	247.5	180.7	147.4
δ. . . . . =	.09	.07	.06	.0543
s. . . . . =	.006413	.00396	.00289	.002357
I. . . . . =	25.65	15.84	11.56	9.428
n. . . . . =	2339	1894	1730	1591
w. . . . . =	2565	1584	1156	942.8
Copper, lbs. . . . . . =	63.73	39.35	28.73	23.44

If it is desired to use metric units (1) should read: Pull in kilograms $= aB^2 \div 24{,}655{,}000$, and (2): $B = nI \div 0.795z$, where B is the flux density in lines per sq. cm., a = area in sq. cm., and z = gap-length in cm.

The foregoing is an abstract from a paper presented at the International Electrical Congress, St. Louis, 1904, by Prof. S. P. Thompson, F.R.S.

(1) and (2) may be combined into the form $P = a(nI \div 2{,}660z)^2$. Mr. C. R. Underhill (E. W. & E., 5-20-05) states that this expression is at best incomplete and offers the following formula: Pull at any point l_a, $P = a(nI \div 2{,}660z)^2 + al_aP_c(nI - k) \div 0.4L(10{,}000 - k)$, where L = length of winding or solenoid, l_a = distance plunger has entered the coil, from end of winding, P_c and k having the values given in the succeeding paragraph on "Solenoid and Plunger."

Solenoid and Plunger. The ampere-turns (nI) required to produce a pull of P lbs. on a plunger of Swedish iron may be calculated from the following formulas, which are due to C. R. Underhill (E. W. & E., 5-13-05): $nI = [10{,}000P - k(P - P_c)] \div P_c$; $A = 0.01\sqrt{nI}$; $d = 0.1128\sqrt[4]{nI}$; where P_c = pull in lbs. on 1 sq. in. of plunger section when $nI = 10{,}000$, A = area of section in sq. in., and k = an empirically determined factor. P_c and k are to be determined from the following formulas which have been derived by the compiler from curves in the original article: $P_c = (102.73 + 0.2105L) \div (1.684 + L)$; $k = (66{,}000 - 3{,}000L) \div (L + 18)$, where L = length of plunger (and generally that of solenoid) in in.

In calculating, add 10% to P desired, and the range through which it will be practically uniform will $= 0.5L$.

Example: For a pull of 30 lbs. over 5 in., $P = 30 \times 1.1 = 33$; $L = 5 \times 2 = 10$ in.; $P_c = 8.973$; $k = 1{,}285.7$; $nI = 33{,}334$; $A = 1.83$ sq. in.; $d = 1.523$ in. From an examination of the data employed by Mr. Underhill the compiler has deduced the following formula, which is much simpler and sufficiently accurate: $nI = 96P(L + 1)$.

CONTINUOUS-CURRENT DYNAMOS.

Connections and Flow of Current. Series-wound dynamo: Armature—field magnets—external circuit—armature.

Shunt-wound dynamo: Armature—$\left\{\begin{array}{l}\text{field magnets}\\ \text{external circuit}\end{array}\right\}$—armature.

Compound-wound dynamo, short shunt:

Armature—$\left\{\begin{array}{l}\text{series magnet coils—external circuit}\\ \text{shunt magnet coils}\end{array}\right\}$—armature.

Compound-wound dynamo, long shunt.

Armature—series coils—$\left\{\begin{array}{l}\text{external circuit}\\ \text{shunt magnet coils}\end{array}\right\}$—armature.

(In the brackets the current divides between the paths in the upper and lower lines inversely as their respective resistances.)

Efficiencies of Dynamos. Let E = E.M.F. in volts; I = armature current in amperes; e = volts at terminals of dynamo; i = amperes in external circuit; i_s = amperes in shunt coils; EI = total watts; ei = useful watts in external circuit; R_1 = armature resistance; R_2 = series-coil resistance; R_3 = shunt-coil resistance; r = resistance of external circuit (all resistances in ohms). N = r.p.m.; η_e = electrical efficiency $= ei \div EI$; η_m = commercial efficiency $= ei \div 746 \times$ H.P. Then for magneto and separately excited dynamos, $\eta_e = e \div E = r \div (r + R_1)$; for series-wound dynamos, $\eta_e = e \div E = r \div (r + R_1 + R_2)$; for shunt-wound machines, $\eta_e = ei \div EI = i^2r \div (i^2r + i_s^2R_3 + I^2R_1)$; for compound-wound, short-shunt dynamos, $\eta_e = ei \div EI = i^2r \div [i^2(r + R_2) + i_s^2R_3 + I^2R_1]$; for compound-wound, long-shunt dynamos, $\eta_e = ei \div EI = i^2r \div [i^2r + I^2(R_1 + R_2) + i_s^2R_3]$.

The Armature. Let n_1 = number of coils on armature and n_2 = number of turns per coil; then, the number of active conductors for a ring armature, $n_0 = n_1n_2$,—for a drum armature, $n_0 = 2n_1n_2$. The E.M.F. $= \Phi n_0 s 10^{-8} \div 60$, where s is the number of revolutions per minute. The cross-section of the armature iron, $a = \Phi \div B$, where B = 10,000 to 16,000 lines per sq. cm. (65,000 to 100,000 lines per sq. in.) for soft charcoal-iron discs, the lower

values for multipolar machines. (For the air-gaps take only about 40% of these values.)

In order to avoid sparking Kapp states that B should equal or exceed $2{,}500(nI)_1 \div [(nI)_1 - (nI)_2]$ for ring armatures (for drum armatures take 60% of value for ring armature), where $(nI)_1$ is the number of ampere-turns required to overcome the reluctance of the air-gap, and $(nI)_2$ is the number of back ampere-turns of the armature. [$(nI)_2$ = no. of conductors included by one pole-face × current strength in amperes.]

The current in the armature sets up a magnetization opposed to that of the field magnets, and the effective field is the resultant of the two.

The external diam. of armature, $d_e = k\sqrt[3]{EI \div N\lambda}$ (J. Fisscher-Hinnen), where λ = length of armature $\div d_e$. For ring armatures, $k = 11.5$ when d_e is in cm. and $= 4.6$ when d_e is in inches; $\lambda = 0.5$ to 1.4. N = r.p.m. For drum armatures, $k = 10$ (d_e in cm.), and $= 4$ (d_e in in.); $\lambda = 0.75$ to 2.8.

The diam. of hole in armature disc, $di = (0.7 \text{ to } 0.8)d_e$ for ring armatures and $(0.3 \text{ to } 0.6)d_e$ for drum armatures. The peripheral speed, s, should not exceed 50 ft. per sec. (15 meters) for small armatures, and 80 ft. per sec. (25 m.) for large armatures. (In exceptional cases it may reach 100 ft. per sec. as for steam turbine generators.)

The length of an armature, $l, = (1.05 \text{ to } 1.2)\dfrac{a}{d_e - di}$ for smooth-surfaced armatures. For toothed armatures, d_e is the diameter at the bottom of the teeth. The cross-section of the armature conductors is determined by allowing 600 to 800 circular mils per ampere. To find the diameters of cotton-covered wires, add the following values to the diameters of bare wires:

Gauge.	Single-covered.	Double-covered.
0 to 10	0.007 in.	0.014 in.
10 " 18	.005 "	.01 "
18 and upwards	.004 "	.008 "

In order to avoid eddy currents armatures are made up of discs of sheet metal (0.015 to 0.025 in. thick) which are insulated from each other by sheets of tissue-paper, rust, or by japanning their surfaces. A sheet of good insulating paper-board is inserted at about every half inch of length and open spaces are left about every two inches to provide for ventilation.

The loss in watts due to eddy currents $= (14.5 \text{ to } 16.5) \times k(Btp)^2 \times 10^{-14}$, where k = cu. cm. of iron in the core, t = thickness of discs in mm., and p = no. of periods per sec.

Armatures, when adequately ventilated in order to avoid injurious heating, and running at peripheral speeds of from 30 to 50 ft. per sec., require from 5 to 7 sq. cm. (0.75 to 1.08 sq. in.) of external surface from which to radiate the heat of each watt wasted therein. (Kapp.)

The permissible rise in temperature (40° to 50° C., or 75° to 90° F.) is t (in degs. C.) $= 85.25W \div S(1 + 0.0305s)$; t (in degs. F.) $= 153.45W \div S(1 + 0.0305s)$, where W = watts lost in armature, S = outside surface of armature in sq. in. and s = peripheral speed in ft. per sec.

In order to avoid fluctuations of E.M.F. and the sparking due to self-induction, the number of coils on the armature should never be less than 30, and as much larger as is consistent with the design. The E.M.F. between two consecutive segments of the commutator should not exceed $(45 - 0.2I)$ volts for currents under 100 amperes, and 20 to 25 volts for heavier currents. The radial depth of the windings on an armature should not exceed one-tenth of the core diam. so that the distance between the core and the pole-faces may be as small as possible. The core should be well insulated from the windings by means of press-board, canvas, etc.

In driving the armature, each conductor opposes the motion by a resistance, or "drag," F (in kilograms) $= lBI \div 9.81 \times 10^6$, where l = length of conductor in cm., and B = induction per sq. cm. F, lbs. $= lBI \div 11{,}303{,}000$, where l is in inches and B in lines per sq. in. I = current in amperes).

The wires should therefore be secured against motion relative to the core surface. In small armatures the frictional resistance of the windings is sufficient, and in toothed armatures the teeth provide backing for the wires. The coils must also be held in place against the action of cen-

trifugal force by bands of German silver or steel wire which are tightly wound around the exterior of the coils in the plane of revolution, secured by soldering or brazing, and insulated from the coils by a layer of mica from 0.012 to 0.025 in. thick. The band wires are from 0.04 to 0.08 in. in diam. and the bands are from 0.6 to 1.2 in. wide. The clearance between the bands and the pole-faces should be from 0.08 to 0.2 in.

Field Magnets. In order that a magnetic flux of Φ_a lines may pass through the armature core there must be a certain number of ampere-turns on the field magnets. The dynamo is to be considered as a closed magnetic circuit through whose several parts (armature core, air-gaps, magnet cores, and yoke) the lines of force flow. For each separate part, $\mathfrak{F}=\Phi\mathfrak{R}$, and, as $\mathfrak{F}=0.4\pi nI$, the ampere-turns $nI=0.7958\Phi\mathfrak{R}$. If l is the length of the mean path of the lines of force in each part in cm., and a the cross-section of each part in sq. cm., then, for the air-gaps, $nI=0.7958Bl$; for iron, $nI=0.7958Hl$, where $B=\Phi\div a$, $H=B\div\mu$, and $\mu=1$ for air. In the following table B is given as a function of $0.7958H=H'$, so that $nI=H'l$, i.e., H' is the number of ampere-turns required to force B lines through 1 cm. length of iron.

Ampere-turns for 1 cm. length of mean path of lines of force (H').

B. per sq. cm.	B. per sq. in.	Sheet metal.	Cast steel.	W. I	C. I.
2,000	12,900	0.35	0.65	0.5	3
4,000	25,800	.75	1.3	1	6.5
6,000	38,700	1.1	2.1	1.7	18
7,000	45,150	1.25	2.65	2	31
8,000	51,600	1.4	3.25	2.35	48
9,000	58,050	1.6	4	2.8	72
10,000	64,500	1.75	5	3.4	97
11,000	70,950	2	6.5	4	133
12,000	77,400	2.7	8.6	5	176
13,000	83,850	4	12	7	232
14,000	90,300	6.5	18	12	
15,000	96,750	12	26.8	21	
16,000	103,200	21	40.6	40	
17,000	109,650	40	58	72	
18,000	116,100	71	93	120	

The above values are for first-quality American metals. (Sheldon.) To find the number of ampere-turns per inch of length, multiply values in table by 2.54. The value of μ may be found from table, it being equal to $0.7958B\div H'$.

For high densities such as are found in the teeth of sheet-metal armature discs,

B per sq. cm. .... =	19,000	20,000	21,000	22,000	23,000
H' per cm. .... =	100	184	320	800	1,450

Calculation of the Ampere-turns of a Dynamo. Armature: Φ_a, a_a, and B_a are determined by the design of the armature; l_a is approximately measured from the dimensions of the core discs, and, the value of H'_a corresponding to B_a being taken, $(nI)_a=H'_al_a$.

If the armature is toothed, a special calculation is necessary; a_t is then the cross-section of the iron in the teeth before one pole-face and should be of such an area that B_t is about 19,000 per sq. cm.

Air-gaps: $\Phi_{\text{air}}=\Phi_a$; $l_{\text{air}}=2\delta$, where δ = distance from armature core to pole-face; $a_{\text{air}}=\lambda b$, where λ and b are respectively the length of the arc and the breadth of the pole-faces. $B_{\text{air}}=\Phi_{\text{air}}\div a_{\text{air}}$ and $(nI)_{\text{air}}=1.5916B_{\text{air}}\delta$.

Field:—Not all of the flux in the field magnets passes through the armature, a part being lost through leakage between the poles. This stray field amounts to from 10 to 50% of the total flux and the field flux must therefore be accordingly greater than that required by the armature. The number of lines of force in the field, $\Phi m=c\Phi_a$, where c has the following values:

Types of Field Magnets.	Capacities of Dynamos in Kilowatts.						
	1	10	100	300	500	1,000	2,000
Upright bipolar, yoke at top, $c=$	1.65	1.45	1.3	1.2			
Same,—yoke at bottom . . . $c=$	1.45	1.28	1.2				
Vertical double magnet (Manchester)	1.8	1.55	1.4				
Radial outward multipolar	1.5	1.32	1.25	1.2	1.18	1.16	1.15
Same, but with inner poles	1.4	1.3	1.22	1.18	1.15	1.12	1.1
Axial multipolar	2	1.7	1.55	1.45	1.4	1.35	1.3

The sectional area, a_m, is calculated in accordance with the permissible B_m, which for C.I. is from 5,000 to 10,000 lines per sq. cm. (32,000 to 64,000 per sq. in.), and for W. I. and steel is from 10,000 to 16,000 per sq. cm. (65,000 to 103,000 per sq. in.). Then, $(nI)_m = H'_m l_m$.

If the cores, yoke, and pole-pieces are of different materials, a separate calculation of the (nI) for each should be made and their sum taken. On account of the reaction of the armature current upon the field the latter is weakened and it is therefore necessary to add from 7 to 15% to the number of ampere-turns. This amount may be approximately calculated by the following formula of Kapp: Let g = the shortest distance between two pole-pieces; then, $(nI)_g = n_0 I g \div \pi(d_e + 2\delta)$, where n_0 = No. of active conductors on the armature, d_e = external diam. of armature in cm. and δ = air-gap between armature core and pole-face in cm.

Finally, the total number of ampere-turns required in the field magnets, $nI = (nI)_a + (nI)_{air} + (nI)_m + (nI)_g = i_m n_m$.

In series machines $i_m = I$ or a fractional part thereof. In shunt machines i_m is determined by the loss permissible in the coils for excitation. The mean length of one turn L_m (in meters) is previously calculated; the resistance, r_m is calculated with regard to the permissible drop, e_m, and $r_m = e_m \div i_m$. The cross-section of the magnet wire in sq. mm. is then, $a_w = L_m nI \div 55 e_m$.

The current density in the field coils should not exceed 2 amperes per sq. mm. (1,300 amp. per sq. in.). In shunt machines from 20 to 40% of the field resistance is used for regulation.

Kapp states that from 10 to 16 sq. cm. of outside coil surface (1.5 to 2.5 sq. in.) is necessary to radiate the heat of each watt lost in the coils. The rise in temperature (25° to 35° C.) t (C°) = (280 to 320)W ÷ surface in sq. cm. = (43.4 to 49.6)W ÷ surface in sq. in. Also, t(F.) = (78 to 89)W ÷ surface in sq. in. W = No. of watts.

Fields should be massive, compactly designed with well fitted joints, and in large sizes should be of W. I. or steel as C. I. requires too great a weight of copper. A circular section should be preferably adopted, sharp edges and corners being avoided, as they tend to increase the leakage. Sparking may be decreased by so boring and adjusting the pole-pieces that the tips are farther distant from the armature-core than are the points midway between the tips.

Eddy currents in pole-pieces may be avoided by slitting the faces in planes at right angles to the axis of rotation of armature, or by constructing the pole-pieces of sheet-iron laminations.

The Commutator segments should be from 0.25 to 0.4 in. thick, made of cast or hard-drawn copper, and insulated from each other by thicknesses of from 0.025 to 0.04 in. of mica. The segments should have a length of about 1.25 in. for each 100 amperes of current, when copper brushes are used. When carbon brushes are employed, length should be from 1.8 to 2.5 in. per 100 amperes.

Brushes. Copper brushes should have a surface of contact with the commutator of from 0.0055 to 0.007 sq. in. per ampere, brass brushes from 0.008 to 0.01 sq. in. per ampere and carbon brushes froom 0.018 to 0.038 sq. in. per ampere. Each brush should cover about 1.5 segments and should be from 1.5 to 2 in. in width, excepting in small machines, where lesser widths are used.

Armature Shafts should possess unusual stiffness in order that vibration may be avoided. Diam., $d = c\sqrt[3]{\text{H.P.} \div N}$, where c = 16 to 23 when d is in cm. and 6.3 to 9 when d is in inches.

The Weight of a Continuous-Current Dynamo in lbs. $= 386K^{\frac{2}{3}}$, where K = output in kilowatts at 1,000 r.p.m. (Fisscher-Hinnen). About

0.2 of this weight is in the armature If the dimensions of a dynamo are multiplied by m, the output will be increased $m^{2.5}$ times, with equal circumf., speed of armature, equal heating, etc. (Kapp.)

The Design of Large Multipolar Dynamos. The following matter, abridged from a series of articles by H. M. Hobart, M. I. E. E. in Technics (London, Jan. to July, 1904), will serve as an illustration of the methods employed in the design of large continuous-current generators. A 400-kilowatt machine (550 volts, 730 amperes) with 8 poles (100 r.p.m.) is taken as an example. E.M.F. $=4TNM\times10^{-8}$ (1), where T = no. of armature turns in series between + and − brushes, N = cycles per sec. or periodicity of reversals of flux in armature core, M = magnetic flux linked with coils in armature. The armature has a multiple-circuit winding, there being 8 paths through it for the current. The external diam. $D=230$ cm. The polar pitch $\tau=\pi\times230\div8=91$ cm. Gross length of armature between flanges, $\lambda_g=40$ cm. There are 8 ventilating ducts, each 13 mm. wide, and 10% of the net length is taken up by insulation. ∴ Net length between flanges, $\lambda_n=27$ cm. The mean length of one armature turn (lap winding) $=3\tau+2\lambda_n=327$ cm. Total number of slots $=264$, and, as there are 6 conductors per slot, the total number of face conductors $=264\times6=1{,}584$, and the total number of turns $=1{,}584\div2=792$. Turns in series between brushes $=792\div8=99$. Total length of conducting circuit between brushes $=327\times99=32{,}400$ cm. Cross-section of one conductor $=2.4$ mm. $\times$ 13 mm. $=0.312$ sq. cm. Total cross-section between brushes (8 conductors in parallel) $=0.312\times8=2.5$ sq. cm. Armature resistance at 60° C. $=32{,}400\times0.000002\div2.5=0.026$ ohm. Voltage drop in armature $=IR=730$ amp. $\times0.026$ ohm $=19$ volts. Drop at brushes $=2$ volts (ranges from 1.2 to 2.8 volts). Assumed drop in compound winding $=3$ volts. Total drop in machine $=24$ volts. Internal voltage $=550+24=574$ volts. $N=(100\div60)\times(8\div2)=6.67$, and $T=99$; substituting these values in (1), $M=21{,}800{,}000$ lines.

Core loss due to hysteresis and eddy currents: Watts per kilogram of weight $=2.54\times$ periods $\times$ kilolines per sq. cm. $\div100$ (2). If the internal diam. of armature disc $=140$ cm., gross area of disc $=\frac{\pi}{4}(230^2-140^2)=26{,}100$ sq. cm. Area of one slot (3.3 cm. deep $\times$ 1.23 cm. wide) $=4.06$ sq. cm. Area of 264 slots $=4.06\times264=1{,}100$ sq. cm. ∴ Net area of disc $=26{,}100-1{,}100=25{,}000$ sq. cm. Volume of iron in core $=25{,}000\times27(=\lambda_n)=675{,}000$ cu. cm. $=5{,}250$ kgs. The core is 42 cm. deep below the slots, consequently the cross-section of core $=42\times27=1{,}135$ sq. cm., but, as the field flux divides as it enters the core and flows both to the left and right, twice this value, or 2,270 sq. cm., = area of core, and the flux density in core will then be $21{,}800{,}000\div2{,}270=9{,}600$ lines, or 9.6 kilolines. The core loss in watts per kg. from (2) $=2.54\times6.67\times9.6\div100=1.7$, or for the entire core $=5{,}250\times1.7=8{,}900$ watts.

Watts per square decimeter of external cylindrical surface of armature: The over-all length of armature may be taken as $L=\lambda_g+0.7\tau=104$ cm. Surface $=\pi DL=\pi\times230\times104=75{,}000$ sq. cm. $=750$ sq. dm. The loss in the copper of armature conductors $=I^2R=730^2\times0.026=13{,}100$ watts, and the total armature loss $=13{,}100+8{,}900=22{,}000$ watts. Watts per sq. dm. $=22{,}000\div750=29.4$, for which value the rise in temperature will not exceed 30° C.

The M.M.F. corresponding to 9,600 lines per sq. cm. $=4$ ampere-turns per cm. of length for sheet iron. (This value for English metal is much higher than that given in preceding table of values for H' of American sheet iron.) The length of path in armature per pole $=42$ cm. ∴ $42\times4=168$ = ampere-turns per coil = M.M.F. for armature core.

Tooth density and the corresponding M.M.F.: $\tau=91$ cm.; arc of pole-face $=61$ cm.; ∴ pole-arc $=0.67\tau$. There are $264\div8=33$ teeth per pole, 67% of which (22.2) lie below the mean pole-arc. Allowing 10% for "spread" of flux, the total number of teeth through which the flux passes $=24.4$ Diam. of armature at the bottom of slots $=223$ cm., and circumference at same diam. $=700$ cm. $700\div264=2.66$ cm. = tooth pitch at bottom of slots. Width of slot is taken $=1.23$ cm., leaving width of tooth $=1.43$ cm. 24.4 teeth $\times1.43=34.8$ cm. at roots. $34.8\times\lambda_n$ or $27=940$ sq. cm. = area of magnetic circuit at roots of teeth for one pole, and the apparent flux density $=21{,}800{,}000\div940=23{,}200$ lines per sq. cm. This apparent

density must not be employed, but a corrected one which varies according to the ratio of the slot width (a) to the tooth width (b). In this case $a \div b = 1.43 \div 1.23 = 1.16$, and by interpolating in the following table the corrected density is found to be 21,800 lines per sq. cm., requiring 640 amp.-turns per cm., or, as length of tooth = 3.3 cm., 2,100 amp.-turns per coil for the teeth.

Apparent Density.	Corrected Density.			
$a \div b =$	0.5	0.75	1	1.25
18,000	17,400	17,700	18,000	18,300
20.000	18,800	19,200	19,500	20,000
22,000	20,000	20,400	20,700	21,300
24,000	21,000	21,500	22,000	22,400
26,000	22,000	22,600	23,000	23,400
28,000	23,000	23,600	24,000	24,500
30,000	23.700	24,600	25,000	25,500

Air-space or gap: Area of pole-face = pole-are $\times \lambda g = 61 \times 40 = 2,440$ sq. cm. Average pole-face density $= 21,800,000 \div 2,440 = 8,900$ lines per sq. cm. Ampere-turns per coil $= 0.795 \times$ average density $\times$ length of gap in cm. $= 0.795 \times 8,900 \times 0.9 = 6,400$.

Magnet cores and yoke: Cores may be of east-steel, W. I., sheet metal, or C. I.; yokes of C. I. or cast steel,—occasionally of sheet metal. Densities for large machines are kept around 14,000 to 15,000 lines per sq. cm. for cast steel and at about 16,000 for W. I. In smaller machines lower values are taken. The flux for the cores and yoke must be greater than that in the air-space and the armature (or account of leakage or dispersion of the lines of force when leaving the poles), and the armature flux must be therefore multiplied by a leakage factor, or, as it is called by Prof. S. P. Thompson, a dispersion coefficient, which ranges from 1.1 in very large machines to 1.25 in small and compactly designed ones. In this example it is taken at 1.13 and the flux in field is therefore $21,800,000 \times 1.13 = 24,600,000$ lines. The core density is then $24,600,000 \div 1,630 = 15,100$ lines for cast steel, the core being 45.5 cm. in diam. and having an area of 1.630 sq. em. The yoke is of cast steel and is designed for 9,000 lines per sq. cm., and has therefore a total sectional area of 2,772 sq. cm., but as the flux divides after leaving the core and flows to the right and left, this value is seen to be twice the actual cross-section, which is 1,386 sq. em.

The length of the path of flux in the magnet core is 50 cm. and that for the yoke and pole-shoe is 73 cm. ($= \frac{1}{2}$ of the total length of path in the yoke between two consecutive cores). The number of amp.-turns per cm. length of eore at 15,100 lines = 28, and for total length of 50 cm. = 1,400 amp.-turns. The amp.-turns per cm. of yoke length at 9,000 lines = 6 or for total length of 73 cm. = 440 amp.-turns.

Total ampere-turns per eoil for 574 volts, at no load:

Armature core below the slots	168
" teeth	2,100
Air-space	6,400
Magnet core	1,400
Yoke	440
Total	10,508

The direct demagnetizing effect of the armature winding when a current is flowing is very considerable and increases the more the brushes are displaced from the mechanical neutral point. This effect may be closely calculated from the formula: Amp.-turns per field coil to overcome demagnetizing component of the armature field $= 0.0175 IPT_a$, where I = amperes per turn in armature coil, T_a = armature turns per pole, and P = percentage of polar pitch by which the brushes are set in advance of the neutral point. In this example, $I = 730 \div 8 = 91$ amp., $T_a = 99$, and, if brushes are set ahead 15 segments of the eommutator, $P = 15 \times 100 \div 99 = 15.2\%$, and $0.0175 IPT_a = 2,400$ amp.-turns.

The distortional component of the field set up by the armature current may be taken at 10% of the total armature field per pole = 730 amp. $\times$

99 turns $\times 0.10 \div 8 = 900$ amp.-turns. Therefore for 550 terminal volts (574 volts internal) at full load are required:

Amp.-turns for saturation at no load........	10,508
" to counteract demagnetization...	2,400
" " " distortion.........	900
Total.............................	13 808 per pole

(In a two-pole dynamo, if the brushes are set at the mechanical neutral point, i.e., at right angles to the direction of the flux, the current in the armature will produce a flux at right angles to that of the fields and tending to distortion of the same. If the brushes are set at 90° from the neutral point, the effect of the armature current is purely one of demagnetization, the flux it produces being directly opposed to the field flux. The brushes being generally set at some intermediate point, it will be seen that both distortion and demagnetization have to be considered). At no load and 550 volts the saturation turns required $= (550 \div 574) \times 10{,}428 \times 0.93 =$ 9,300 amp.-turns, where 0.93 is a factor which approximately allows for the bending of the no-load saturation curve. The shunt coils must therefore have 9,300 amp.-turns at all loads, and the series coils at full load $13{,}728 - 9{,}300 = 4{,}428$ amp.-turns.

Space factor in winding:—in armatures with voltages up to 1,000 the insulation thickness between the copper and iron should range from 1.15 mm. to 2 mm.,—or, for the present design, say a slot lining 0.4 mm. thick and insulation wrapped around coil of about 0.6 mm. The double-covering of cotton on the conductors may be considered as adding 0.3 mm. to the diam. of the bare wire. The ratio of actual copper section to the slot section is called the space factor and should be as high as possible, thereby increasing the output of the machine. This factor is higher the fewer the number of slots and is lower the smaller the diam. of conductors used. Space factors for armatures range from 0.3 to 0.5 for round wires and from 0.36 to 0.6 for conductors of rectangular cross-section. Space factors for field coils range from 0.4 to 0.65, a good average value being 0.5. The value 0.65 is used for series coils with large conductors of rectangular cross-section which are wound edgewise.

Calculation of field coils:—Space factor taken at 0.5 for both coils. The length allowable for winding = 40 cm. (i.e., 50 cm. minus the thickness of flanges, pole-shoe, etc.). Dividing this length in proportion to the number of ampere-turns gives a length of 28 cm. for the shunt coil and 12 cm. for the series coil. At full load (coil at 60° C.) 10% of the shunt excitation is wasted in an adjusting rheostat in series with the coils. This reduces the voltage from 550 to 500 volts, or 62.5 volts for each of the 8 coils. Allowing 1 cm. for clearance, the internal diam. of coil = 46 cm., and assuming radial depth to be 4 cm., the external diam. will be 54 cm., and the mean length of one turn (a) will be 1.58 meters. The watts per shunt coil at 60° C. $= 0.000176 a^2 b^2 \div k$, where k = kgs. of copper per coil and b = amp.-turns per coil (= 9,300). Cross-section of shunt coil = $28 \times 4 = 112$ sq. cm., which, multiplied by the space factor (0.5) = cross-section of copper in coil = t = 56 sq. cm. Cu. cm. of copper in coil = $56 \times 1.58 \times 100 = 8{,}900$, and, as 1 cu. cm. weighs 0.0089 kg., the kgs. of copper in one shunt coil = 79. Substituting these values in above formula, the watts per shunt coil = 480.

The external cylindrical surface of coil = 48 sq. dm., and the watts per sq. dm. therefore = 10, which allowance will not permit a rise in temperature of more than 40° C.

Size of wire in shunt coils:—Amps. per coil = watts ÷ volts per coil = $480 \div 62.5 = 7.7$ amp. Turns per coil = amp.-turns ÷ amps. = $9{,}300 \div 7.7 =$ 1,210. Cross-section per turn = $t \div$ No. of turns $= 56 \div 1{,}210 = 0.0462$ sq. cm. Current density $= 7.7 \div 0.0462 = 167$ amp. per sq. cm. Diam. of bare wire = 2.42 mm. Watts in 8 coils = 3,840. Watts in shunt rheostat = 380. ∴ Total watts for shunt = 4,220. Copper in 8 coils = 630 kgs.

Series coils:—These are placed at the end of core nearest the armature. Winding length = 12 cm. Turns = 4,420 amp.-turns ÷ 730 amp. = 6 turns. (In this particular machine 210 amp. are diverted through a shunt in parallel with the series winding so that turns $= 4{,}420 \div 520 = 8.5$.) The series coils may have a higher current density than the shunt coils, and, if this is taken at 180 amp. per sq. cm., the cross-section of the series turns = $730 \div 180 = 4.05$ sq. cm. This may be in the shape of a rectangular section

(4 cm. $\times$ 1.01 cm.) and wound edgewise. Mean length of 1 turn = 158 cm. Weight of copper in one coil = 6 turns $\times 158 \times 4.05 \times 0.0089 = 34.17$ kgs., or 273.36 kg. for 8 coils. Resistance of 8 coils in series at 60° C. $= 8 \times 6 \times 158 \times 0.000002 \div 4.05 = 0.00374$ ohm.

Watts lost in the 8 coils, at 60° C. $= 730^2 \times 0.00374 = 1{,}993$.

Reactance voltage:—When a coil carrying a current arrives at and passes the brush, the direction of the current is suddenly reversed. This change should take place sparklessly and the winding should be so designed that the reactance voltage due to the decreasing current at the moment of commutation will be as small as possible at full load, the brushes being set at the neutral point. Reactance voltage $= 12.566e\left(\frac{Q}{B}\right)\left(1+0.15\frac{\tau}{\lambda n}\right)$, where e = average voltage per coil ($= 550 \div 99 = 5.5$ volts), Q = amperes in conductors per em. of periphery of armature $\left(=\frac{730}{8}\times\frac{1{,}584}{230\pi}=200 \text{ amp.}\right)$, B = average flux density per sq. cm. of cylindrical surface of armature [$=(8\times 21{,}800{,}000)\div(28\times 230\times\pi)=8{,}600$ lines], and $\tau \div \lambda_n$ = ratio of polar pitch to net length of armature core ($= 99 \div 27 = 1.49$).

The reactance voltage, consequently, is 2.42 volts for this machine, which is low enough to permit a practically sparkless commutation. The brushes should be held against the commutator by a pressure of about 0.1 kg. per sq. cm., and the loss in watts due to brush friction = 0.1 kg $\times$ section of brushes in sq. cm. $\times 0.3 \times$ peripheral speed of commutator in meters per second $\times 9.81$, where 0.3 = coeff. of friction for carbon brushes (= 0.2 for eopper brushes). The current density in brushes ranges from 4 to 12 amp. per sq. cm.,—average = 6.

The IE loss at commutator in watts = total armature eurrent $\times$ volts dropped at brushes (1.2 to 2.8,—average, 2).

Efficiency:—The following is a tabulation of the several losses of energy in the generator at full load:

(a) Core loss in armature	8,900	watts	(constant)
(b) I^2R " " "	13,100	"	(variable)
(c) Brush contact loss	1,460	"	"
(d) Brush friction loss	540	"	(constant)
(e) Friction loss at bearings, estimated	3,000	"	"
(f) Loss in shunt coils	3,840	"	"
(g) " " series "	1,993	"	(variable)
Total losses	32,833	watts	

Output $= 730 \times 550 = 401{,}500$ watts. Total generated $= 401{,}500 + 32{,}833 = 434{,}333$ watts. Efficiency at full load $= 401{,}500 \div 434{,}333 = 92.5\%$. At half-load, losses $= a+d+e+f+\frac{1}{2}(b+c+g) = 24{,}560$ watts. Output = 200,-750 watts, and total generated $= 200{,}750 + 24{,}560 = 225{,}310$ watts. Efficiency at half-load $= 200{,}750 \div 225{,}310 = 89\%$.

Cost of manufacture: The factory cost of generators of this class is proportional to the product of the diameter of the armature by the "equivalent length of one armature turn over the end connections," which latter may be taken $= \lambda_g + 0.7\tau$. The factory cost then $= KD(\lambda_g + 0.7\tau)$, K being a function of voltage and of the type of machine. For 6 and 8 pole dynamos of 250 volts, K may be taken at \$0.30, and for 500 volts at \$0.265 to \$0.28. (These values are for material and labor eosts and for methods of manufacture obtaining in England.)

The output and speed being decided upon, a series of calculations should be made, the diameter of armature being so chosen that the peripheral speed will vary from 10 to 15 meters per sec. and the total ampere-turns per pole on the armature varying from 4,000 to 10,000. From these designs a choiee may be made which will be the best compromise on such points as cost, speed, and reaetance voltage, all of which should be as low as possible.

For a two-circuit winding on a multipolar dynamo armature, where one pair of brushes is used, No. of face conductors = No. of poles $\times$ (winding piteh ± 2).

CONTINUOUS-CURRENT MOTORS.

These are generally designed on the same lines as are dynamos of similar types. The revolutions of the armature develop an E.M.F. which is op-

posed to the impressed E.M.F. and which is called the counter electromotive force. Let E = E.M.F. applied at the terminals of motor, e = counter E.M.F., and R = resistance of motor armature. Then, $I=(E-e)\div R$; total watts, $W=EI=E(E-e)\div R$; useful watts, w, $=eI=e(E-e)\div R$; $W=w+I^2R$ (or watts lost in heating), and the efficiency $=w\div W=e\div E$.

Torque = mechanical power in ft.-lbs. ÷ angular velocity. Let $\omega=2\pi\times$ revs. per sec. = angular velocity, T = torque; then, ωT = mechanical power in ft.-lbs. per sec. eI = electrical power of the armature in watts. H.P. = $\frac{\omega T}{550}=\frac{eI}{746}$, and $eI=2\pi nT\times\frac{746}{550}=8.52nT$, where n = revs. per sec. $e=nm\Phi 10^{-8}$, where m = No. of conductors on the periphery of armature and Φ = flux. T at 1 ft. radius $=m\Phi I\div(8.52\times10^8)$. If r = resistance of armature, $I=\frac{(E-e)}{r}$, and T (at 1 ft.) $=m\Phi\left(\frac{E-e}{r}\right)\div(8.52\times10^8)$. R.p.m. $=e\times60\times10^8\div m\Phi$.

Rheostats for Motors. If a motor at rest were directly connected to a source of current, the mains would be short-circuited through the armature and the abnormal current flowing would speedily burn up the armature coils. It is necessary, therefore, to introduce a starting resistance into the armature circuit so that only a moderate current will flow through the armature at the beginning of its motion. As the speed (and consequently the counter E.M.F.) increases, the current strength decreases, and the resistance may be lowered gradually, by steps, and when full speed is attained it may be cut out of the circuit altogether. The following table gives the resistance and current-carrying capacity of several metals used in rheostat coils:

B.W.G.	Galvanized Iron. Ohms per ft.	Galvanized Iron. Amp.	German Silver: Ohms per ft.	German Silver: Amp.	Platinoid. Ohms per ft	Platinoid. Amp.	Manganin. Ohms per ft.
8	0.00266	28	0.00566	19	0.008	13.5	0.0093
10	.00366	21	.00833	14	.0123	10	.0133
12	.006	16	.0127	11	.019	7.7	.021
14	.0117	10	.0203	7	.032	4.7	.0363
16	.016	7.5	.0333	5	.05	3.5	.0553
18	.029	4.5	.0583	3	.089	2.2	.1013
20	.041	3.5	.115	2.2	158	1.5	.1446
22	.0883	2	.18	1.5	.262	.95	.3133
24	.144	1.5	.29	1	.423	.7	.5

Resistance coils should be wound according to the following table, which gives the sizes for maximum rigidity and energy dissipation:

B. W. G.	Inner diam. of Spiral in inches.	Approx. length Coil in inches.
8	1	27
9 to 11	0.875	22
12 " 14	.75	18
15 " 16	.625	14
17 " 19	.5	11
20 " 24	.375	8

A starting resistance should be so designed that the momentary increase of current due to cutting out a section of same does not exceed a certain predetermined amount.

$$0 - r - \overset{1}{0} - R_1 - \overset{2}{0} - R_2 - \overset{3}{0} - R_3 - \overset{4}{0} - R_n - \overset{n}{0} \text{———} \quad \text{Current flow.}$$

In the above diagram r is the armature resistance, R_1, R_2, R_3, R_n are the sectional resistances of the rheostat included between the segments 1, 2, 3, 4, n. Let the E.M.F. of supply $=E$; i = current in armature at full load; I = permissible momentary current, and let $I\div i=k$. The resistance R_1 between segments 1 and 2 should then be $=(k-1)r$, $R_2=(k-1)kr$, $R_3=(k-1)k^2r$, and $R_n=(k-1)k^{n-1}r$.

In order for the motor to start, the total resistance in the circuit $(=r+R_1+R_2+R_3\ldots+R_n)$ must be less than $E\div i$. To avoid arcing between the segments no section should have a drop of over 35 volts, and

if such a section should occur in the calculation it should be divided into two or more sections, none of which have a drop exceeding 35 volts. For motors using about 50 amperes on full load I may be taken as equal to $i+10$ amp. For much smaller motors $I=\frac{i}{2}$ for the first section and $=\frac{i}{4}$ on the remainder. If the full-load current exceeds 50 amp. the momentary rise $(I-i)$ should not exceed $0.2i$.

Example: Rheostat for a 15 H.P. motor on a 220 volt circuit. $E=220$; $i=15\times746\div220=50$ amp. Resistance of armature, $r=0.15$ ohm. $I=i+10=60$ amp. $I\div i=k=1.2$. $E\div i=4.4$ ohms. The sum of all the resistances $=rk^n$, where $n=$ No. of sections in the rheostat $=E\div i$. $\therefore$ $0.15k^n=4.4$, and, as $k=1.2$, $n=18$ (18.5 exactly) sections and the resistance of each section may be calculated from the previous formulas. If the rheostat is designed to start the motor, say on half-load, $i=25$; $I=i+10=35$; $I\div i=k=1.4$, whence $1.4^n=58.66$, and $n=12$ sections. $E\div i=$ total resistance $=8.8$ ohms. The several sections would have the following values:

$r=$	0.15	ohm	$R_7=$	0.4518	ohm
$R_1=(k-1)r=$	.06	"	$R_8=$	.6325	"
$R_2=R_1k=$	.084	"	$R_9=$	.8855	"
$R_3=R_2k=$	.1176	"	$R_{10}=$	1.2397	ohms
$R_4=$	.1646	"	$R_{11}=$	1.7355	"
$R_5=$	.2305	"	$R_{12}=$	2.4297	"
$R_6=$	.3227	"	Total $=$	8.5	"

As n is a fraction over 12, the remaining 0.3 ohm $(8.8-8.5)$ may be added to R_{12}. (Condensed from an article by F. H. Davies, in Technics, April, 1904.)

ALTERNATING CURRENTS.

Definitions. Alternating currents are those which periodically pass through a regular series of changes both in magnitude and direction. Usually the magnitude increases with a certain regularity from zero to a maximum, decreases with the same regularity to zero, and then similarly to a maximum in the opposite direction and finally to zero again. When a current has experienced such a series of changes (0 to $+$max., to 0, to $-$max., to 0) it is said to have completed one cycle. (Symbol $\frown$.) There are two alternations in one cycle. The time taken to accomplish one cycle is called a period and the number of cycles completed in one second is called the frequency, or periodicity. The frequency of an alternating current dynamo $=pN\div60$, where $p=$ number of pairs of poles, and $N=$ r.p.m.

The ideal curve of an alternating current and E.M.F. is a sinusoid, or curve of sines, and is the one assumed for purposes of theoretical discussion, but commercial alternators do not generate strictly sinusoidal pressures.

Referring to Fig. 29, E' at any point $=E_{\text{max.}}\sin 2\pi ft$, where $f=$ frequency and $t=$ time in seconds. Also, $I=I_{\text{max.}}\sin 2\pi ft$.

Effective Values. One ampere of alternating current is a current of such instantaneous value as to have the same heating effect in a conductor as one ampere of direct or continuous current. Heating varies as I^2 and, therefore, in an alternating current whose instantaneous values vary, the heating effect is proportional to the mean of the squares of the instantaneous currents, or, $I^2=Im^2\div2$. The effective value, therefore is $I=Im\div\sqrt{2}$, and the effective E.M.F., $E=Em\div\sqrt{2}$. The average current, $I_{\text{av.}}=2Im\div\pi$, and the average E.M.F., $E_{\text{av.}}=2Em\div\pi$. The ratio of the effective E.M.F., and the average E.M.F. $=\frac{Em}{\sqrt{2}}\div\frac{2Em}{\pi}=1.11$ (for sinusoidal E.M.F.s) is called the form factor. (The subscript m indicates maximum.)

Phase. When the maximum and zero values of E and I occur at the same instant, the current and E.M.F. are said to be in phase. When the current attains its maximum and zero values at a time later than when the corresponding values of the E.M.F. occur, it is said to be out

of phase with the E.M.F., or to lag behind the E.M.F. When maximum and zero values are reached at an earlier time, the current is said to lead the E.M.F. The distance between any two corresponding ordinates of current and E.M.F. may be measured and expressed in degrees and is called the angular displacement, or phase difference. This angle is represented by ϕ.

An alternator giving a single pressure wave of E.M.F. to a two-wire circuit is called a single-phase current generator. One giving pressure to two distinct circuits (each a single phase), the phases being 90° apart, is a two-phase, or quarter-phase generator. A three-phase machine

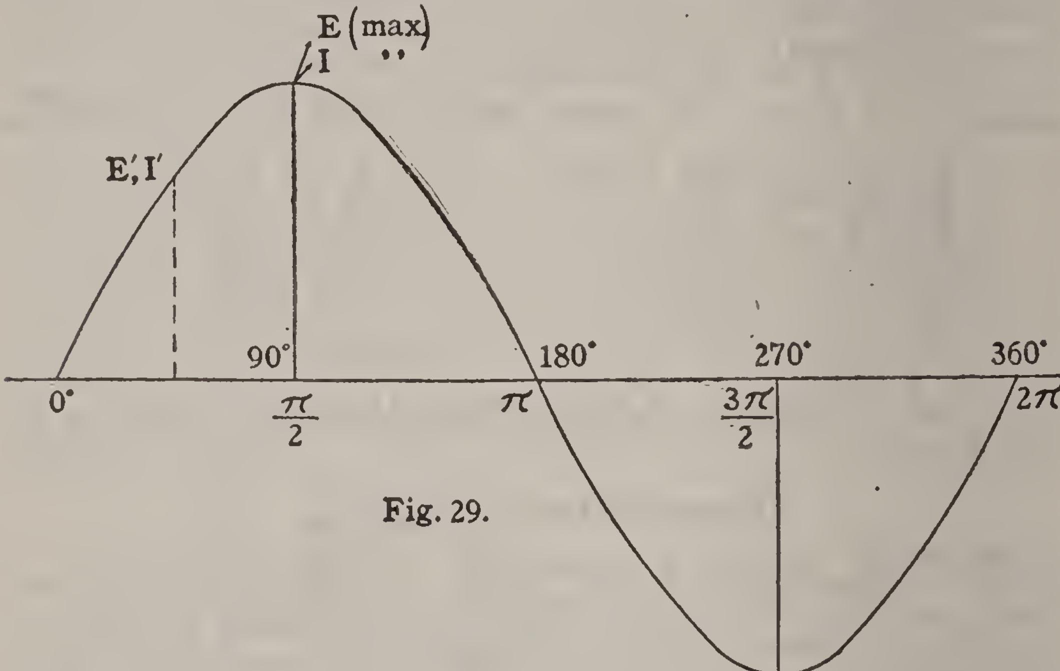


Fig. 29.

theoretically has three two-wire circuits, the maximum positive pressure on any one circuit being displaced from each of the pressures in the other two circuits by 120°, but, as the algebraic sum of the currents in all three circuits (if balanced) $=0$, the three return wires of the circuits may be dispensed with.

Power in Alternating-Current Circuits. The power, P, in an alternating circuit depends on E, I, and ϕ, and is thus expressed: $P=EI\cos\phi$. Cos ϕ is called the power factor, it being the number by which the apparent power, or volt-amperes (EI), must be multiplied in order to obtain the true power. When E and I are in phase, $\phi=0$ and $\cos\phi=1$.

Self-Induction:—Impedance, Reactance, and Inductance. A current flowing in a conductor sets up a magnetic field around it; conversely, when there is an increase or decrease of the number of lines of force cut by a conductor, a current is induced in it, and in alternating circuits it is necessary to consider these self-induced currents.

When the rate of change of value of the current strength is greatest (at 0) the self-induced E.M.F. is a maximum, and when lowest (at peak of the sine curve) the E.M.F. is a minimum; consequently, the phase of the self-induced E.M.F. differs from that of the impressed E.M.F. by 90°, or is at right angles to it.

Let an alternating current of I amperes flow through a circuit having a resistance of R ohms and an inductance (self-induction) of L henrys. To maintain the current flow through R requires an effective E.M.F., $E_r=RI$. The effective value of the E.M.F. of self-induction, E_s, will be $=-2\pi fLI$, the minus sign indicating that it is an opposed, or counter E.M.F. As E_r and E_s are at right angles to each other they are not to be added, but are to be taken as two sides of a triangle, the hypothenuse of which is the impressed E.M.F., E; whence, $E=\sqrt{E_r^2+E_s^2}=\sqrt{(IR)^2+(2\pi fLI)^2}$, and $I=E\div\sqrt{R^2+(2\pi fL)^2}$. $\sqrt{R^2+(2\pi fL)^2}$ is called the

impedance, or apparent resistance, and $(2\pi fL)$ the reactance, both being expressed in ohms (Fig. 30).

As E_r is the part of the impressed E.M.F. which sends the current through the conductor (Es being that required to neutralize the self-induction), the current must be in phase with it, and I is therefore always displaced 90° from Es. I and E_r lag behind E by an angle (ϕ) whose cosine $= E_r \div E$.

The inductance of a coil on the field of a generator is: L (in henrys) $= \Phi n I 10^{-8}$, where Φ is the total flux from one pole, n the number of turns in coil, and I the amperes of current in coil.

Capacity. Any two conductors separated by a dielectric (i.e., insulating substance) constitute a condenser. In practice this term applies to a collection of thin sheets of metal separated from each other by thin sheets of insulation, every alternate sheet of metal being connected to one terminal of the apparatus and the intervening leaves of metal to the other terminal. The function of a condenser is to store up electrical energy. If a continuous E.M.F. be applied to a condenser, a current will flow,—large at first, but gradually diminishing until the metal sheets have been charged to an electrostatic difference of potential equal and opposed to that of the E.M.F. applied. The capacity of a condenser is numerically equal to the quantity of electricity with which it must be charged in order to raise the difference of potential between its terminals from zero to unity. A condenser whose potential is raised 1 volt by the charge of 1 coulomb has a capacity of 1 farad.

The capacity in microfarads of a condenser $= C = 0.000225\frac{Ank}{t}$, where A = area of dielectric between two metal leaves, in sq. in.; n = number of sheets of dielectric; t = thickness of dielectric in mils; k = specific inductive capacity of the dielectric.

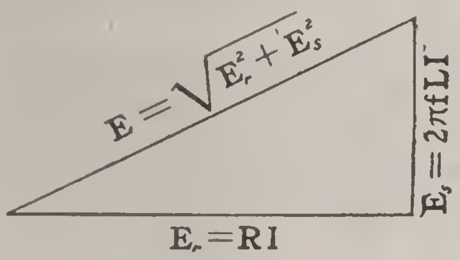


Fig. 30.

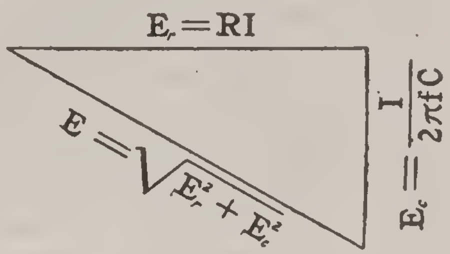


Fig. 31.

Values of k:—Glass, 3 to 7; ebonite, 2.2 to 3; gutta-percha, 2.5; paraffin, 2 to 2.3; shellac, 2.75; mica, 6.6; beeswax, 1.8; kerosene, 2 to 2.5.

If a sinusoidal E.M.F., E, of frequency, f, be impressed on a condenser, the latter will be charged in $\frac{1}{4f}$ seconds, discharged in the next $\frac{1}{4f}$ seconds and charged and discharged in the opposite direction in equal succeeding intervals. Max. voltage, $Em = E\sqrt{2}$; max. quantity, $Qm = EC\sqrt{2}$; quantity per second $= 4fQm = 4fEC\sqrt{2}$ = average current, $I_{av.}$, and, as the effective current, $I = \frac{\pi}{2\sqrt{2}} I_{av.}$, $I = 2\pi fCE$, and $E = \frac{1}{2\pi fC} I$. $\frac{1}{2\pi fC}$ is called the capacity reactance and is analogous to $2\pi fL$.

Circuits containing Resistance and Capacity. In this case the impressed voltage, E, must be considered as being made up of E_r,—which sends the current through the resistance, R,—and E_c, which balances the counter pressure of the condenser and which is 90° in phase behind the current. $E_r = RI$, and $E_c = \frac{1}{2\pi fC} I$. $\therefore$ Impressed E.M.F, $E = \sqrt{E_r{}^2 + E_c{}^2}$ or $I = \frac{E}{\sqrt{R^2 + \left(\frac{1}{2\pi fC}\right)^2}}$. (See Fig. 31.)

Circuits containing Resistance, Inductance, and Capacity. This is the most general case. The counter E.M.F. due to self-induction $=2\pi fL$, and leads the current by 90°. The E.M.F. of capacity reactance $=\frac{1}{2\pi fC}$, and lags behind the current by 90°. These two E.M.F.'s being 180° apart, the resultant reactance is their numerical difference and the general equation is: $I=E\div\sqrt{R^2+\left[2\pi fL-\frac{1}{2\pi fC}\right]^2}$. The quantity within the brackets indicates an angle of lag, if positive, and an angle of lead, if negative. If $2\pi fL=\frac{1}{2\pi fC}$, then $I=\frac{E}{R}$. This condition prevailing, resonance is said to exist, as, at one instant, energy is being stored in the field at the same rate it is being given to the circuit by the condenser, and at another instant, energy is being released from the field at the same rate as it is being stored in the condenser.

Combinations of Condensers. If condensers are connected in series their combined capacity, $C=\frac{1}{\frac{1}{C_1}+\frac{1}{C_2}+\frac{1}{C_3}\cdots+\frac{1}{Cn}}$. If $C_1, C_2, \ldots Cn$ are equal capacities, $C=\frac{C_1}{n}$.

If connections are in multiple, $C=C_1+C_2+C_3+\ldots Cn$, and if $C_1=C_2=C_3=Cn$, $C=nC_1$.

Combinations of Impedances. If several impedances are to be arranged in series they should be represented by the hypothenuses of triangles whose horizontal sides represent the resistances and vertical sides the reactances. The resultant impedance is then represented by the hypothenuse of the triangle whose base = sum of the resistance horizontals of the separate triangles and whose height = sum of the reactance verticals, or, resultant impedance $=\sqrt{\Sigma R^2+\Sigma[2\pi fL]^2}$.

If the impedances are in parallel, find their reciprocals or admittances. Take any two admittances at their proper phase angle and construct a parallelogram. The diagonal will be the resultant of these two admittances in direction and value. This resultant may be similarly combined with a third admittance, etc. The reciprocal of the final resultant admittance will then be the combined impedance desired and the direction of the final diagonal will represent the resultant phase.

ALTERNATING-CURRENT GENERATORS.

Alternators are either single-phase or poly-phase (i.e., more than one phase,—usually two or three). For low potentials the field is stationary, the armature revolving, while for high potentials the field is made to rotate, the armature being fixed. The latter may have a field of radial poles each of which is of opposite polarity to its neighbor, or, it may be of the inductor type, in which both field and armature coils are stationary, the rotating part being an iron mass called the inductor. This inductor (which carries no wire) has pairs of soft-iron projections termed inductors which are magnetized by the current flowing in a fixed annular field coil which surrounds but does not touch the inductor. The surrounding frame is provided with radial internal projections which correspond to the inductors in number and size, and upon which are wound the armature coils. As the inductors revolve the flux linked with the armature coils varies from a maximum to a minimum, but its direction is not changed, as the annular field coil gives a constant direction of field.

Two-Phase Generator. In a two-phase system of winding, if two coils and 4 conductors are used, each coil generates a pressure of E volts between the two wires leading from it and there is no connection between the two coils. If three wires are used, connected as shown in Fig. 32, the E.M.F.'s between the wires are as indicated in the diagram. (E and I in the figures are taken as the effective E.M.F.'s and currents.)

A monocyclic generator (for lights chiefly, but carrying a certain motor load) is a single-phase machine to which is added on the armature a so-

called "teazer" winding of a section sufficient to carry the motor load, and with turns enough to produce a voltage equal to one-quarter of that

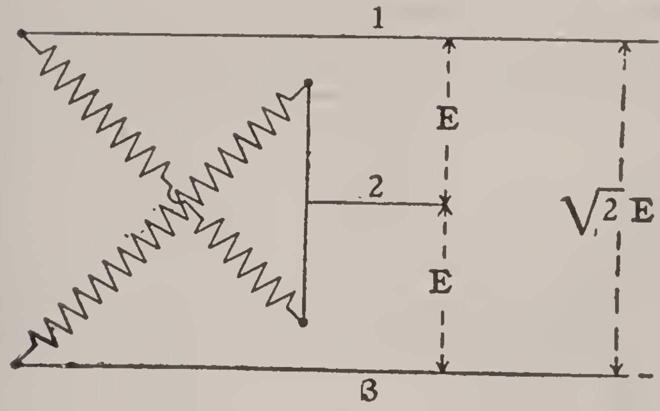


Fig. 32.

of the regular winding and lagging 90° behind same. One end of the teazer winding is connected to the middle of the regular winding and the

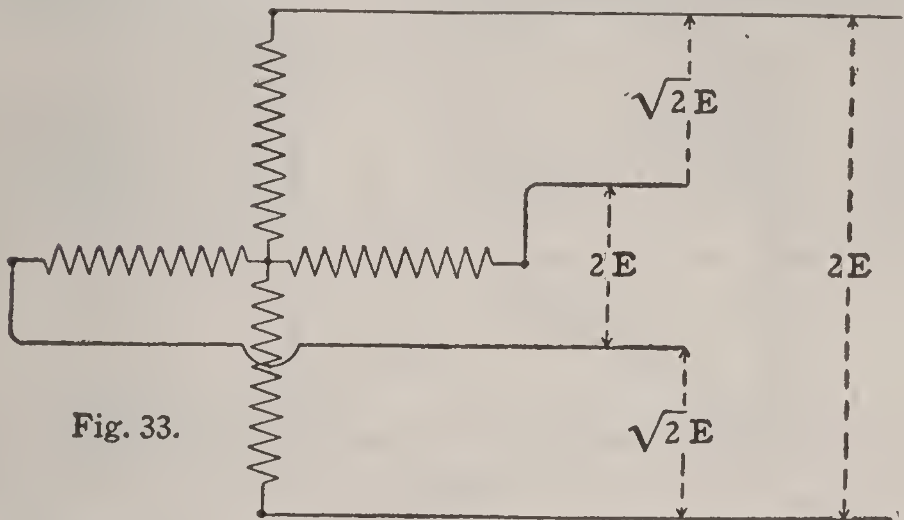


Fig. 33.

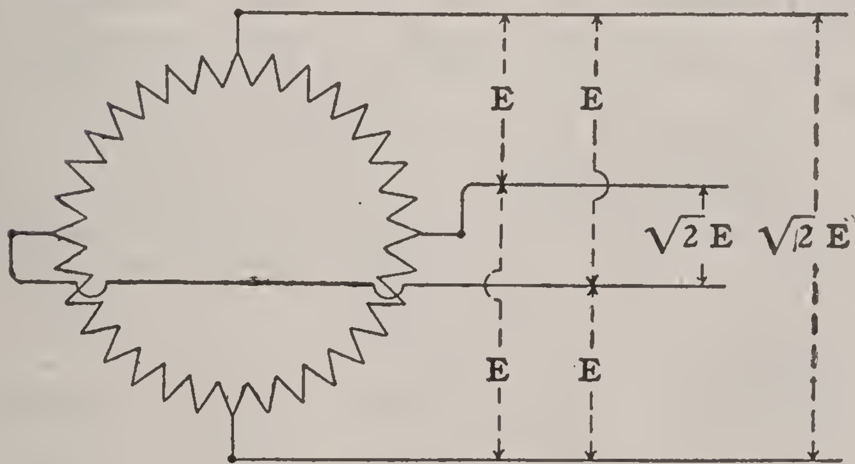


Fig. 34.

other to a third line-wire. A three-terminal induction motor is used, which is either connected directly or through a transformer.

Four-Phase, or Quarter-Phase. See Figs. 33 and 34 for the two

styles of connections. The current in each line in Fig. 33 $= I$, and in each line of Fig. 34 $= I\sqrt{2}$.

Three-Phase (Figs. 35 and 36). Fig. 35 shows the Y or "star" connection, the current in each line being I. Fig. 36 shows the Δ (delta) or mesh connection, the current in each line being $I\sqrt{3}$.

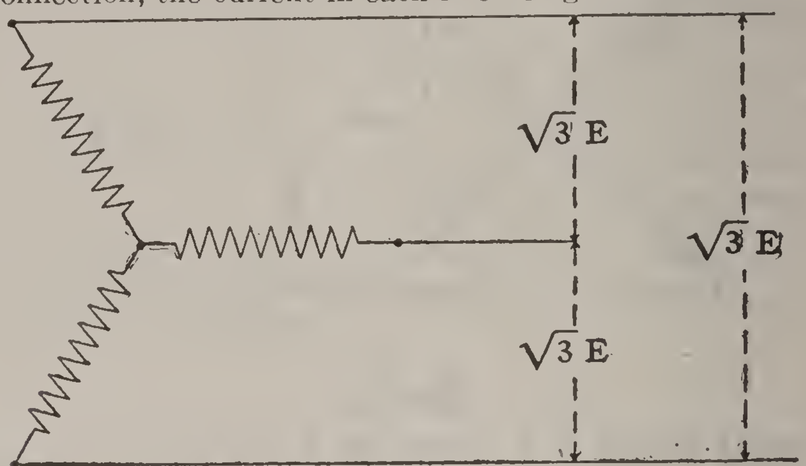


Fig. 35.

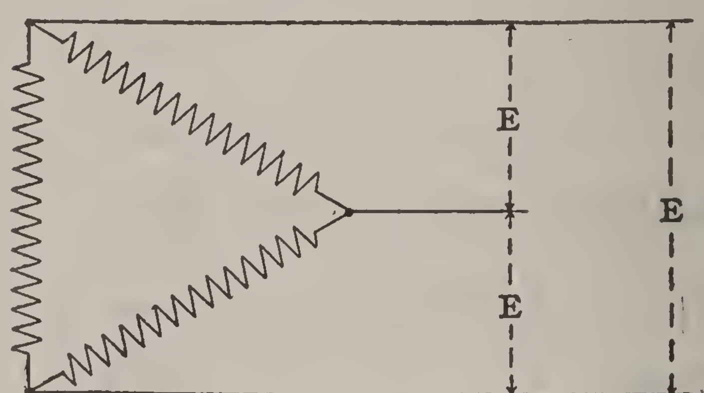


Fig. 36.

E.M.F. Generated. $E_{av} = 2p\Phi n \frac{N}{60} 10^{-8}$, where p = number of pairs of poles, Φ = flux per pole in maxwells, N = r.p.m., and n = number of inductors. The effective E.M.F. $= kE_{av}$, where k is the form factor ($=1.11$ for a sine wave). Also, $pN \div 60 = f$, consequently $E = 2.22\Phi nf10^{-8}$.

If the armature winding is all concentrated into one slot per pole, single-phase, this formula is applicable. If, however, the wires are distributed over the surface of the armature in a number of slots the right-hand member of the equation must be multiplied by a distribution constant, k_1, which varies according to the number of slots on the periphery of armature from center to center of two adjacent pole-faces and the fraction of the latter distance which is occupied by slots.

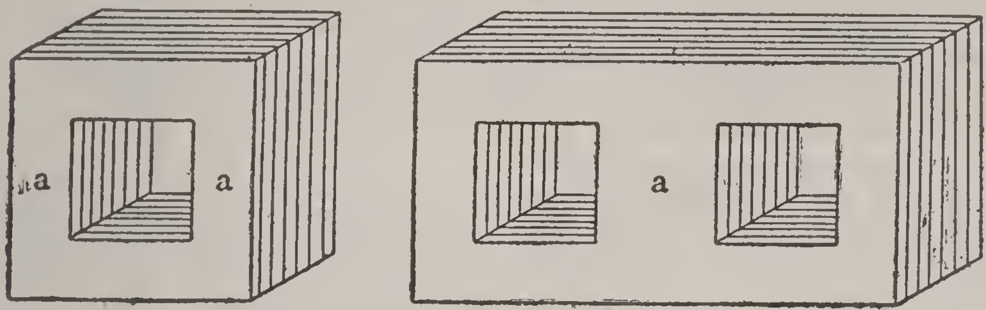
Values of k_1.

Part of polar distance occupied by slots.	1 slot.	2 slots.	3 slots.	many slots.
0.1	1.00	0.996	0.995	0.994
0.2	1.00	.986	.984	.982
0.3	1.00	.972	.967	.962
0.4	1.00	.95	.942	.935
0.5	1.00	.925	.912	.9

TRANSFORMERS.

The transformer is a device for changing the voltage and current of an alternating electric system and consists of a pair of mutually inductive circuits (primary and secondary) or coils interlinked with a magnetic circuit or core. When an alternating voltage is applied to the primary coil an alternating flux is set up in the iron core which induces an alternating E.M.F. in the secondary coil in direct proportion to the ratio of the number of turns of the primary and secondary coils.

The magnetic circuit or core is made up from laminations of sheet iron or steel. Two general types are used: I, the core type, which is built up from laminations, each of which is a rectangle, with a similar but smaller rectangle stamped out from its center. These laminations are bound together with the holes corresponding and coils are wound on two opposite limbs. II, the shell type, which is similarly assembled, but in which each lamination has two rectangular holes stamped out. The coils are wound on the central limb formed by the bridges or cross-pieces between the rectangular holes in the laminations. Laminations are about 0.014 in. thick and are insulated from each other by shellac, tissue paper, etc., in much the same manner as are the discs in armature cores. (See Fig. 37, the coils being wound on the limbs marked *a*.)



I. II.

Fig. 37.

Volts induced in transformer coil, $E=4.44T\,.\,\Phi f 10^{-8}$, where T = total number of turns of wire in series and f = frequency, in cycles per sec.

Eddy current losses:—Watts per cu. cm. of core $=(tfB)^2 10^{-16}$, where t = thickness of each lamination in mils, and B is in lines per sq. cm.

Amperes required to magnetize core to induction $B=\frac{Bl}{1.76\mu T}$, where l = length of magnetic circuit in cms., B = lines per sq. cm., T = No. of turns in primary coil, and μ = permeability of the iron in core.

The current at no load

$$=\sqrt{(\text{magnetizing current})^2+\left(\frac{\text{watts lost in iron}}{\text{primary voltage}}\right)^2}.$$

Transformer Design (abridged from articles by Prof. Thos. Gray, in E. W. & E., April 23 and 30, 1904).

Let a, b, and l be the dimensions in cm. of the cross-section and mean length of the copper link or coil, and a_1, b_1, and l_1 be similar dimensions for the iron link or core. Then, total cross-section of coils $=ab=A$, and cross-section of core $=a_1b_1=A_1$. Volume of iron, $v_1=A_1l_1$, and volume of coils, $v=Al$. (In this discussion the laminations are assumed to be rectangular and the wires as being bent sharply at right angles as they turn the corners of the core.)

For a core transformer, ab = total section of *both* coils. $l=2\left(a_1+\frac{A}{a_1}+a\right)$, and $l_1=2\left(a+\frac{A}{a}+2a_1\right)$. In order that l may be a minimum (assuming

A, A_1, and l_1 to be constants and differentiating), it is found that for this condition $\frac{a_1}{b_1}=\frac{b-a}{b+a}$, and $\frac{a}{b}=\frac{b_1-a_1}{b_1+a_1}$.

For the least total volume of material in both cores and coils it is found necessary that $a^2=\frac{AA_1}{A+A_1}$, $b^2=\frac{A(A+A_1)}{A_1}$, $a_1{}^2=\frac{AA_1}{A+2A_1}$, and $b_1{}^2=\frac{A_1(A+2A_1)}{A}$. If the volumes are to have a definite relative value, let $v_1=nv$, and let the corresponding relative value of the areas be: $A_1=xA$.

When $x=$	0.5	1	1.5	2	3	4	5
$n=$	0.796	1.086	1.286	1.435	1.637	1.77	1.864

Let the induction per sq. cm. of core $=B \sin \omega t$, and the total induction $=A_1 B \sin \omega t$; then, the magnetizing current being small, the amplitude of the applied E.M.F. will (when the transformer is not loaded) be practically equal to that induced by self-induction; consequently, $E=n_1 A_1 B\omega 10^{-8}$, where $n_1=$No. of turns on primary coil.

Let $P=$full load in watts, $I=$square root of the mean square of the full-load current in primary coil, and power factor$=1$. Then, $P=EI\div\sqrt{2}$, or $1.41P=EI$. Let $i=$average current per sq. cm. of coil section. The heat generated in the coils will then be, approximately, $=4i^2v10^{-6}$, assuming the space factor of the coils is 50% (i.e., one-half of coil section is copper), and the working temperature$=80°$ C.

At full load the heat wasted in the coils should equal that lost in the core through hysteresis and eddy currents. This heat, $H=\frac{18B^{1\cdot6}}{10^{11}}$ in watts per cu. cm. per cycle per second.

A certain area of radiating surface, s, must be allowed for the dissipation of the heat of each watt, the total surface being S. For ordinary air-cooled transformers s is taken at 30 sq. cm., and at 20 sq. cm. for transformers immersed in oil or cooled by artificial ventilation. The following equations and values have been derived from the foregoing premises:

$$B^{9\cdot2}=7.866\times10^{55}\times\left(\frac{2\pi}{\omega}\right)^5\left(\frac{g}{s}\right)^8\times\frac{v_1}{v}\times\frac{x_1{}^4x^2}{P^2}\quad(1).$$

$$a=\frac{2\pi}{\omega}\times\frac{g}{s}\times\frac{10^{11}}{36B^{1.6}}\quad(2).\qquad n_1=\frac{10^8E}{xx_1B\omega a_1}\quad(3).$$

Total heat dissipated, $H_1=2\times18B^{1.6}\,v_1\frac{\omega}{2\pi}\,10^{-11}$, in watts per sec. (4).

$\frac{w}{2\pi}=$frequency; $g=$total exposed surface $\times\frac{a}{v_1}=\frac{Sa}{v_1}$; $x_1=\frac{b}{a}$; $x=\frac{A_1}{A}$.

$x=\frac{A_1}{A}=$	0.25	0.5	1	1.5	2	2.5	3.5
$x_1=\frac{b}{a}=$	6	3	2	1.66	1.5	1.4	1.285
$x_2=\frac{b_1}{a_1}=$	1.5	2	3	4	5	6	8
$g=\frac{Sa}{v_1}=$	6.83	4.76	3.46	2.83	2.46	2.22	1.92

(Read thus When $x=1$, $x_1=2$, $x_2=3$, and $g=3.46$, etc.)

Example Core transformer; $P=10{,}000$ watts, $E=3{,}000$ volts, $\frac{w}{2\pi}=100$, $x=1$, and from previous tables $x_1=2$, $\frac{b_1}{a_1}=3$, $\frac{v_1}{v}=n=1.086$, and $g=3.46$. Substituting in (1), (2), (3), and (4), $B=2{,}747$ lines per sq. cm., n_1 (primary) $=853$ turns, $a=10.1$ cm., $b=20.2$ cm., $a_1=8.25$ cm., $b_1=24.75$ cm., $A=A_1$

=204.2 sq. cm., $v=17,600$ cu. cm., $v_1=19,110$ cu. cm., $v+v_1=36,710$ cu. cm., $H_1=218.3$ watts. Efficiency = watts output ÷ watts supplied = $(10,000-218.3)\div 10,000=97.82\%$. If the iron section is taken as one-half that of the coils, $x=0.5$, $\frac{v_1}{v}=0.796$, $x_1=3$, $\frac{b_1}{a_1}=2$, $g=4.76$, $v=15,940$ cu. cm., $v_1=12,700$ cu. cm., $v+v_1=28,640$ cu. cm., $H_1=222.6$ watts, and eff. $=97.74\%$, or a dissipation of but 4 watts more than in the first case, and a reduction in weight of one-third.

For shell transformers, $B^{9.2}=7.866\times 10^{55}\left(\frac{2\pi}{\omega}\right)^5\left(\frac{g}{s}\right)^8\left(\frac{v}{v_1}\right)^7\frac{x_2{}^4}{x^2P^2}$ (5),

$$a_1=\frac{2\pi}{\omega}\times\frac{g}{s}\times\frac{v}{v_1}\times\frac{10^{11}}{36B^{1.6}} \quad (6), \quad \text{and} \quad n_1=10^8E\div x_2B\omega a_1{}^2 \quad (7).$$

$x_2=\frac{b_1}{a_1}$; $g=\frac{Sa_1}{v_1}$. In this case, where (e.g.) $x=1$, and the iron parts correspond to the copper parts in a core transformer, the values of $\frac{b_1}{a_1}$ in the table are used for $\frac{b}{a}$ and similarly those of $\frac{b}{a}$ in table for $\frac{b_1}{a_1}$.

Taking the data of the example given, it will be seen that in this case $\frac{b_1}{a_1}=2$ instead of 3 as for a core transformer. Substituting the various values in (5), (6), and (7), the following values are obtained: $B=2,896$, $n_1=813$, $a=8.22$, $b=24.65$, $a_1=10.07$ $b_1=20.13$, $A=A_1=202.7$. $v=18,890$, $v_1=17,400$, $v+v_1=36,290$, $H_1=216.2$; eff. $=97.84\%$, or substantially the same total volume and efficiency as for the core transformer first considered. If the iron section be made equal to twice the copper section, $B=3,091$, $v=13,060$, $v_1=16,400$, $v+v_1=28,140$ cu. cm., $H_1=226.6$, eff. $=97.73\%$. If iron section = copper section $\times 5$, $B=3,770$, $v=7,632$, $v_1=13,640$, $v+v_1=21,000$, $H_1=260.2$; eff. $=97.4\%$.

When a transformer is in circuit continuously, but loaded for only a few hours in the day a greater all-day average efficiency is obtained by designing the transformer so that the iron heat dissipation is considerably less than that of the coil at full load; the efficiency, however, is smaller, on full load. In this case the right-hand members of (1) and (5) must be multiplied by $\frac{m-1}{m^8}$, and those of (2) and (6) by $\frac{1}{m}$, where m = total heat dissipation ÷ hysteresis dissipation.

If $m=3$ (other data as for shell transformer where $B=2.896$), then, $B=2,195$, $v=21,180$. $v_1=19,510$, $v+v_1=40,690$, $H_{iron}=77.9$ watts, $H_{copper}=155.8$ watts, $H_1=77.9+155.8=233.7$ watts. Eff. $=97.66\%$. The weight is thus increased about 12% and the efficiency lowered by 0.18%. If the load, however, is on only about 6 hours out of the 24, there is a saving of about 600 watt-hours per day.

It is assumed in the foregoing work that the coil and core sections are rectangular. If the iron laminations are rectangular and the wires in the coils are bent in the arc of a circle when rounding the corners of the iron core (which is the most general construction), then, for a core transformer, $\frac{b_1}{a_1}=\frac{b+\left(\frac{\pi}{2}-1\right)a}{b-a}=\frac{b+.57a}{b-a}$; $l=2\left(a_1+\frac{A_1}{a_1}+\frac{\pi a}{4}\right)$; $l_1=2\left(a+\frac{A}{a}+2a_1\right)$

All-Day Efficiency. Let y = No. of hours per day when full load is on; then

$$\text{All-day efficiency}=\frac{\text{Full load}\times y}{\text{Core loss}\times 24+\text{copper loss}\times y+\text{full load}\times y}.$$

Magnetic Densities in Various American Transformers:

For 25 cycles,

$B=9,000$ to 10,000 lines per sq. cm. (60,000–90,000 per sq. in.).

For 60 cycles,

$B=6,000$ to 9,000 lines per sq. cm. (40,000–60,000 per sq. in.).

For 125 cycles,
$B=4,500$ to 7,500 lines per sq. cm. (30,000–50,000 per sq. in.).

Current Densities. Primary coil, 1,000–1,500 circular mils per ampere.
Secondary " 1,200–2,000 " " " "

Insulation between laminations is about 10% of total assembled thickness; ∴ vol. of iron $=0.9\times$ cubic contents.

Economic Design. The best economy of first cost may be obtained by calculating several transformers of the same capacity, but with various ratios of copper to iron, plotting the results and balancing the annual interest on the cost of material saved (labor cost being substantially a constant for a given output) with the cost of the extra watt-hours per year sacrificed by cheapening the construction.

CONDUCTORS.

Copper-Wire Table, A. I. E. E. 20° C.

Gauge. B. & S.	Diameter. Inches.	Area. Circular mils.	Weight. Pounds per ft.	Length. Feet per lb.
0000	0.460	211,600	0.6405	1.561
000	.4096	167,800	.5080	1.969
00	.3648	133,100	.4028	2.482
0	.3249	105,500	.3195	3.13
1	.2893	83,690	.2533	3.947
2	.2576	66,370	.2009	4.977
3	.2294	52,630	.1593	6.276
4	.2043	41,740	.1264	7.914
5	1819	33,100	.1002	9.98
6	.1620	26,250	.07946	12.58
7	.1443	20,820	.06302	15.87
8	.1285	16,510	.04998	20.01
9	.1144	13,090	.03963	25.23
10	.1019	10,380	.03143	31.82
11	.09074	8.234	.02493	40.12
12	.08081	6,530	.01977	50.59
13	.07196	5,178	.01568	63.79
14	.06408	4,107	.01243	80.44
15	.05707	3,257	.009858	101.4
16	.05082	2,583	.007818	127.9
17	.04526	2,048	.006200	161.3
18	.04030	1,624	.004917	203.4
19	.03589	1,288	.003899	256.5
20	.03196	1,022	.003092	323.4
21	.02846	810.1	.002452	407.8
22	.02535	642.4	.001945	514.2
23	.02257	509.5	.001542	648.4
24	.02010	404	.001223	817.6
25	.01790	320.4	.0009699	1,031
26	.01594	254.1	.0007692	1,300
27	.0142	201.5	.0006100	1,639
28	.01264	159.8	.0004837	2,067
29	.01126	126.7	.0003836	2,607
30	.01003	100.5	.0003042	2,287
31	.008928	79.7	.0002413	4,145
32	.00795	63.21	.0001913	5,227
33	.00708	50.13	.0001517	6,591
34	.006305	39.75	.0001203	8,311
35	.005615	31.52	.00009543	10,480
36	.005	25	.00007568	13,210
37	.004453	19.83	.00006001	16,660
38	.003965	15.72	.00004759	21,010
39	.003531	12.47	.00003774	26,500
40	.003145	9.888	.00002993	33,410

Copper-Wire Table—(*Continued*).

Gauge. B. & S.	Weight. Pounds per ohm.	Length. Feet per ohm.	Resistance.	
			Ohms per pound.	Ohms per foot
0000	13,090	20,440	0.00007639	0.00004893
000	8,232	16,210	.0001215	.00006170
00	5,177	12,850	.0001931	.00007780
0	3,256	10,190	.0003071	.00009811
1	2,048	8,083	.0004883	.0001237
2	1,288	6,410	.0007765	.0001560
3	810	5,084	.001235	.0001967
4	509.4	4,031	.001963	.0002480
5	320.4	3,197	.003122	.0003128
6	201.5	2,535	.004963	.0003944
7	126.7	2,011	.007892	.0004973
8	79.69	1,595	.01255	.0006271
9	50.12	1,265	.01995	.0007908
10	31.52	1,003	.03173	.0009972
11	19.82	795.3	.05045	.001257
12	12.47	630.7	.08022	.001586
13	7.84	500.1	.1276	.001999
14	4.931	396.6	.2028	.002521
15	3.101	314.5	.3225	.003179
16	1.950	249.4	.5128	.004009
17	1.226	197.8	.8153	.005055
18	.7713	156.9	1.296	.006374
19	.4851	124.4	2.061	.008038
20	.3051	98.66	3.278	.01014
21	.1919	78.24	5.212	.01278
22	.1207	62.05	8.287	.01612
23	.07589	49.21	13.18	.02032
24	.04773	39.02	20.95	.02563
25	.03002	30.95	33.32	.03231
26	.01888	24.54	52.97	.04075
27	.01187	19.46	84.23	.05138
28	.007466	15.43	133.9	.06479
29	.004696	12.24	213	.0817
30	.002953	9.707	338.6	.103
31	.001857	7.698	538.4	.1299
32	.001168	6.105	856.2	.1638
33	.0007346	4.841	1,361	.2066
34	.0004620	3.839	2,165	.2605
35	.0002905	3.045	3,441	.3284
36	.0001827	2.414	5,473	.4142
37	0001149	1.915	8,702	.5222
38	00007210	1.519	13,870	.6585
39	.00004545	1.204	22,000	.8304
40	.00002858	0.955	34,980	1.047

The table is calculated for a temperature of 20° C. Resistance in international ohms. For resistance at 0° C., multiply values in table by 0.9262; for resistance at 50° C., multiply by 1.11723, and for resistance at 80° C., multiply by 1.23815. The following data were used in computing the table: Specific gravity of copper = 8.89. Matthiessen's standard 1 meter-gram of hard drawn copper at 0° C. = 0.1469 British Association unit (B.A.U.) = 0.14493 international ohm (1 B.A.U. = 0.9866 international ohm.) Ratio of resistivity of hard to soft copper = 1.0226. Temperature coefficients of resistance for 20°, 50°, and 80° C. (cool, warm, and hot) taken as 1.07968, 1.20625, and 1.33681, respectively.

Aluminum Wires at 75° F. (Pittsburgh Reduction Co.).

Gauge. B. & S.	Ohms per 1,000 ft.	Feet per ohm.	Ohms per lb.
0000	0.08177	12,229.8	0.00042714
000	.1031	9,699	.00067022
00	.1300	7,692	.00108116
0	.1639	6,245.4	.0016739
1	.2067	4,637.4	.0027272
2	.2608	3,836.2	.0043441
3	.3287	3,036.1	.0069057
4	.4145	2,412.6	.0109773

Conductivity taken as 60% of that of pure copper. Weight of pure aluminum taken as 167.111 lbs. per cu. ft.

General Formulas for Wiring. (From General Electric Co. literature.) Area of conductor in circular mils $= DWK \div PE^2$; Volts lost in line $= PEM \div 100$; Current in main conductors $= WT \div E$; Weight of copper in line $= AWKD^2 \div PE \times 10^6$; where D = distance of transmission (one way) in feet, W = total watts *delivered* at the end of line, P = per cent loss of W in line, and E = voltage between the conductors at the receiving end of line. A, K, and T are constants having the following values:

	A	K — Per cent, power factor — 100	95	90	85	80
Single-phase	6.04	2160	2400	2660	3000	3380
Two-phase (4 wires)	12.08	1080	1200	1330	1500	1690
Three-phase (3 wires)	9.06	1080	1200	1330	1500	1690

	T — Per cent power factor — 100	95	90	85	80
Single-phase	1	1.05	1.11	1.17	1.25
Two-phase (4 wires)	0.5	.53	.55	.59	.62
Three-phase (3 wires)	.58	.61	.64	.68	.72

K for continuous current = 2160, $T = 1$, $A = 6.04$, and $M = 1$.

Values of M.—Wires 18 in. apart from c. to c.

Gauge: B. & S.	25 Cycles. Power factor in per cent. 95	90	85	80	40 Cycles. Power factor in per cent. 95	90	85	80
0000	1.23	1.29	1.33	1.34	1.52	1.53	1.61	1.67
000	1.18	1.22	1.24	1.24	1.40	1.41	1.48	1.51
00	1.14	1.16	1.16	1.16	1.25	1.32	1.35	1.37
0	1.10	1.11	1.10	1.09	1.19	1.24	1.26	1.26
1	1.07	1.07	1.05	1.03	1.14	1.17	1.18	1.17
2	1.05	1.04	1.02	1.00	1.11	1.12	1.12	1.10
3	1.03	1.02	1.00		1.07	1.08	1.07	1.05
4	1.02	1.00			1.05	1.06	1.03	1.00
5	1.00				1.03	1.01	1.00	
6					1.02	1.00		
7					1.01			
8					1.00			

Gauge: B. & S.	60 Cycles. Power factor in per cent. 95	90	85	80	125 Cycles. Power factor in per cent. 95	90	85	80
0000	1.62	1.84	1.99	2.09	2.35	2.86	3.24	3.49
000	1.49	1.66	1.77	1.95	2.08	2.48	2.77	2.94
00	1.34	1.52	1.60	1.66	1.86	2.18	2.40	2.57
0	1.31	1.40	1.46	1.49	1.71	1.96	2.13	2.25
1	1.24	1.30	1.34	1.36	1.56	1.75	1.88	1.97
2	1.18	1.23	1.25	1.26	1.45	1.60	1.70	1.77
3	1.14	1.17	1.18	1.17	1.35	1.46	1.53	1.57
4	1.11	1.12	1.11	1.10	1.27	1.35	1.40	1.43
5	1.08	1.08	1.06	1.04	1.21	1.27	1.30	1.31
6	1.05	1.04	1.02	1.00	1.16	1.20	1.21	1.21
7	1.03	1.02	1.00		1.12	1.14	1.14	1.13
8	1.02	1.00			1.09	1.10	1.09	1.07
9	1.00				1.06	1.06	1.04	1.02
10					1.04	1.03	1.00	1.00

The values of M in the above table are about true for 10% line loss.

They are reasonably accurate for losses less than 10%, under 40 cycles and close enough for larger losses. If the largest conductors are used at 125 cycles and the loss is greater than 20%, the values should not be used. If the conductors are closer to each other than 18 inches, the loss will be less than that given by the formula, and if very close together, as in a cable, the loss will be that due to resistance only.

For a direct-current 3-wire system, the neutral feeder should have a section equal to one-third of that of the outside wires as obtained from formula. For both alternating and direct current the secondary mains and the house wiring should have the neutral wire of the same area as the outside conductors.

For the monocyclic system (power and lights) calculate the primary circuit as if all the power were transmitted over the outside wires, the size of the power wire to be to either outside wire as the motor load (in amperes) is to the total load in amperes. Secondaries leading to induction motors should all be of the same size as for a single-phase circuit of the same capacity in kilowatts and same power factor. The three lines of a 3-phase circuit should be of equal cross-section.

Power Factor:—When not more accurately determinable, take as follows: Lighting only, 95%; lighting and motors, 85%; motors only, 80%. For lighting circuits using small transformers the voltage at transformer primaries should be 3% higher than the voltage × ratio of transformation. For motor circuits substitute 5% for 3% in the preceding rule.

Examples:—Direct-current circuit, 1,000 110-volt lamps, each taking 0.5 ampere; line loss, 10%; two wires; distance, 2,000 feet.

Circular mils $= 2{,}160 \times 2{,}000 \times (1{,}000 \times 0.5 \times 110) \div (10 \times 110^2) = 1{,}963{,}636$.

Volts drop to lamps $= 10 \times 110 \times 1 \div 100 = 11$ volts.

Three-wire circuit,—220 volts between the outside wires: Area of each outside conductor $= 2{,}160 \times 2{,}000 \times (1{,}000 \times 0.5 \times 110) \div (10 \times 220^2) = 490{,}900$ cir. mils. Area of neutral or third wire $= 490{,}900 \div 3 = 163{,}633$ cir. mils. Volts loss in circuit $= 10 \times 220 \times 1 \div 100 = 22$ volts.

Alternating currents: Two-wire, single-phase; 10 to 1 transformers 2 volts loss in secondary wiring; transformer drop $= 3\%$; loss in primary line to be 5% of the delivered power; efficiency of transformer $= 97\%$. Volts at transformer primaries $= (110 + 2) \times 10 \times 1.03 = 1153.6$.

Watts required by lamps $= 1{,}000 \times 110 \times 0.5 = 55{,}000$. Watts required at primaries $= 55{,}000 \div (0.98 \times 0.97) = 58{,}000$. Cir. mils $= 2{,}000 \times 58{,}000 \times 2{,}400 \div (5 \times 1{,}153.6^2) = 41{,}760$.

Three-phase, 3-wire power transmission, 60 cycles; 3,500 H.P., 5 miles; loss, 10% of delivered power; voltage at motor $= 5{,}000$; power factor of load $= 85\%$. Circular mils $= (5{,}280 \times 5) \times (3{,}500 \times 746) \times 1{,}500 \div (10 \times 5{,}000^2) = 413{,}582$. Two 0000 wires have this area, also four 0 wires. If the latter are used, the drop will be only 73.3% of that when using the larger wires $\left(\frac{1.46}{1.99}\right)$. Per cent loss $= 5{,}280 \times 5 \times 3{,}500 \times 746 \times 1500 \div (4 \times 105{,}592) \times 5{,}000^2 = 9.79\%$ of the delivered power, or 322.6 H.P. loss in the transmission. Volts lost in line $= 9.79 \times 5{,}000 \times 1.46 \div 100 = 715$. Volts at generator $= 5{,}000 + 715 = 5{,}715$. Current in line $= 3{,}500 \times 746 \times 0.68 \div 5{,}000 = 355$ amperes.

Calculations applying to Transmission Circuits. The E.M.F.'s in the various parts of a transmission system may be calculated by means of the following table and the method employed in the example given.

Line Constants. (Wires 18 in. apart.)

Gauge, B. & S.	Wt., lbs.	Diam. mils.	Area, cir. mils.	*R.*	*L.*	*C.*	*i.*	Reactance, *X*, $f=25$	40	60	125.
0000	3,376	460	211,600	.266	1.48	.0102	.0385	.232	.372	.558	1.16
000	2,677	410	167,800	.335	1.52	.00996	.0375	.239	.382	.573	1.19
00	2,123	365	133,100	.422	1.56	.00973	.0366	.245	.392	.588	1.22
0	1,685	325	105,500	.533	1.60	.00949	.0358	.251	.402	.603	1.26
1	1,335	289	83,690	.671	1.63	.00926	.0349	.256	.409	.614	1.28
2	1,059	258	66,370	.845	1.66	.00909	.0342	.261	.417	.625	1.30
3	840	229	52,630	1.067	1.70	.00883	.0333	.267	.427	.641	1.33
4	666	204	41,740	1.346	1.73	.00863	.0326	.272	.435	.652	1.36
5	528	182	33,100	1.700	1.77	.00845	.0319	.278	.445	.667	1.39
6	419	162	26,250	2.138	1.81	.00827	.0312	.284	.455	.682	1.42
7	332	144	20,820	2.698	1.84	.00809	.0305	.289	.462	.693	1.44
8	263	128	16,510	3.406	1.88	.00793	.0295	.295	.472	.708	1.48

Weight given in lbs. per mile of wire; R=ohms per mile of conductor; L=inductance in millihenrys per mile; C=capacity in microfarads, of two wires, each one mile in length; i=charging current of line of two wires ($f=60$, $E=10.000$ volts)$=2\pi fCE10^{-6}$; X=reactance$=2\pi fL10^{-3}$. Impedance, $Z=\sqrt{R^2+X^2}$.

Let it be required to transmit 2,700 H.P. over a 3-phase circuit 10 miles in length, the power being generated at 1,000 volts, raised through a step-up transformer to 10,000 volts for transmission along the line, and reduced to 1,000 volts at the receiving end by a step-down transformer. Transformer efficiencies=97.5%; copper loss in each, 1%; core or hysteresis loss, in each, 1.5%; reactance=3.5%; magnetizing current=4%. Loss in transmission=15%, 10 of which is in line; power factor=0.85. Voltage between any branch and the common center of system$=E\div\sqrt{3}=10{,}000\div\sqrt{3}=5{,}774$. Energy delivered by each wire$=2{,}700\times746\div3=671{,}400$ watts. Apparent energy per branch$=671{,}400\div0.85=790{,}000$ watts. Current in each wire$=790{,}000\div5{,}774=136.8$ amperes. Drop in each wire=10% of 5,774=577.4 volts. Resistance of each wire$=577.4\div136.8=4.22$ ohms, or 0.422 ohms per mile, which is the resistance of a 00 wire; consequently, three 00 wires will carry the load. Reactance of 10 miles single conductor$=0.588\times10=5.88$ ohms. Inductance for 10 miles$=10\times1.56=15.6$ millihenrys. Charging current for each line, for 10 miles$=.0366\times10=0.366$ amp. Power factor being 0.85, the inductance factor$=\sqrt{1-0.85^2}=0.52$.

To find the E.M.F. at generator and the distribution of current when full load is on, the entire system may be considered at 10,000 volts for convenience in calculation.

$$\text{Impressed E.M.F.}=\sqrt{\Sigma\,(\text{energy E.M.F.'s})^2+\Sigma\,(\text{Induction E.M.F.'s})^2}.$$

Commencing with the secondary circuit, working back and tabulating the steps, the following is obtained:

		Energy E.M.F.	Inductive E.M.F.	Current.
Secondary Circuit:				
Energy, E.M.F.$=5{,}774\times0.85$	=	4,909		
Inductive E.M.F.$=5{,}774\times0.52$	=		3,003	
Current, in amperes	=			136.8
Step-down Transformers:				
Resistance loss, IR=1% of 5,774	=	58		
Reactance " IX=3.5% of 5,774	=		202	
Hysteresis " =1.5% of 136.8	=			2.05
		4,967	3,205	138.85
Line:				
Resistance loss, $IR=138.85\times4.22$	=	586		
Reactance " $IX=138.85\times5.88$	=		817	
		5,553	4,022	138.85
(Volts at terminals of step-up transformers$=\sqrt{5{,}553^2+4{,}022^2}=6{,}857$.)				
Step-up Transformers:				
Resistance loss, IR=1% of 6,857	=	69		
Reactance " IX=3.5% of 6,857	=		240	
Hysteresis " 1.5% of 138.85	=			2.08
		5,622	4,262	140.93

Volts at generator$=\sqrt{5{,}622^2+4{,}262^2}=7{,}055$ volts, or, reduced by 10.1 ratio, =705.5 volts, for one branch. The total generator E.M.F. would then be $705.5\times\sqrt{3}=1{,}222$ volts, or total volts at generator=122.2% of volts at secondaries of receiving transformers, and the power factor of the entire circuit is $1{,}000\div1{,}222=0.818$.

Inductance for Parallel Copper Wires, Insulated. L per 1,000 feet per wire$=0.01524+0.14\log\frac{d}{r}$; L per 1,000 ft. of the whole circuit for a

3-phase line $=0.02639+0.2425 \log \frac{d}{r}$, where L is in millihenrys, d and r being respectively the distance between centers of wires and radius of wire, both measured with the same unit.

Capacities of Conductors. Lead-protected cables: Microfarads per 1,000 ft. of length $=0.007361K \div \log \frac{D}{d}$. Single overhead conductors, with earth return: Microfarads per 1,000 ft. $=0.007361 \div \log \frac{4h}{d}$.

Each of two parallel, bare aerial wires: Microfarads per 1,000 ft. $=0.003681 \div \log \frac{d_1}{r}$. In the above, D=diam. of cable outside of insulation, d=diam. of conductor, d_1=distance between wires, c. to c., h=height above ground, r=radius of wire, K=specific inductive capacity of insulating material. D, d, d_1, h, and r should all be measured by the same unit.

Heating of Conductors. Insulated parallel wires: Diam. in inches $=0.0147\sqrt[3]{I^2}$ (Kennelly). Bare wires: Diam. in mils $=45\sqrt[3]{I^2 \div (T-t)}$, where I=current in amperes, T=temp. of wire, and t=temp. of air, both in degs. F.

Carrying Capacity of Interior Wires and Cables (A. I. E. E.).

B. & S. Gauge.	Rubber-covered.	Weather-proof.
14	12	16
12	17	23
10	24	32
8	33	46
6	46	65
4	65	92
2	90	131
0	127	185
000	177	262
0000	210	312

Circular mils.	Rubber-covered.	Weather-proof.
400,000	330	500
600,000	450	680
800,000	550	840
1,000,000	650	1000
1,500,000	850	1360
2,000,000	1050	1670

The capacities are in amperes. No smaller wire than No. 14 to be used.

Rubber covering to be $\frac{3}{64}$ in. thick for No. 14 to No. 8, $\frac{1}{16}$ in. for No. 7 to No. 2, $\frac{5}{64}$ in. for No. 1 to 0000, $\frac{3}{32}$ in. for No. 0000 to 500,000 cir. mils., $\frac{7}{64}$ in. up to 1,000,000 cir. mils, and $\frac{1}{8}$ in. above 1,000,000 cir. mils. Weather-proof coverings must have the same thicknesses, the inner coating to be fireproof and 0.6 of the total thickness.

Insulation Resistance (National Code). The wiring in complete installations must have an insulation resistance $=\left(\frac{20{,}000{,}000}{\text{amperes flowing}}\right)$ in ohms.

Fuses. Fuses for 5 amperes and less should be 1.5 in. long, and 0.5 in. should be added for each additional 5 amperes. Round wire should not be used for over 30 amp.,—above that, use a flat strip. Fusing current $=ad^{\frac{3}{2}}$, where d=diam. in inches and a is a constant having the following values: copper, 10,244; aluminum, 7,585; platinum, 5,172; iron, 3,148; tin, 1,642; lead, 1,379; 2 lead+1 tin, 1,318.

Amperes.	Diameter in Inches. Copper.	Iron.	Tin.	Lead.
1	.0021	.0047	.0072	.0081
10	.0098	.0216	.0334	.0375
50	.0288	.0632	.0975	.1095
100	.0457	.1003	.1548	.1739
200	.0725	.1592	.2457	.276
300	.095	.2086	.322	.3617 (Preece.)

ELECTRIC LIGHTING.

Arc Lamps. 45 to 60 volts, 9.6 to 10 amp., 2,000 candle-power (nominal); 45 to 50 volts, 6.8 amp., 1,200 candle-power (nominal). Enclosed arcs require 80 volts, 5 amperes; carbons burn from 100 to 150 hours. Alternating-current arc lamps require 28 to 30 volts and 15 amperes.

The mean spherical candle-power (c.-p.) is the mean of that over a sphere of which the light is the center and equals, approximately, $\frac{H}{2}+\frac{M}{4}$, where H is the horizontal c.-p. and M the maximum c.-p. (40° below horizontal for a direct-current arc). The continental unit of light is the hefner, or 0.88 candle-power.

Clear-glass globes cut off 10% of the illumination, ground-glass globes from 35 to 50%, and opal globes from 50 to 60%.

Incandescent Lamps, usually 16 c.-p., require from 3 to 3.5 watts per c.-p. and have a life of 800 to 1,000 hours. They should not, however, be used over 600 hours, as their efficiencies decrease during use. The most economical point at which to renew a lamp (i.e. the "smashing" point) may be found as follows:

Hours lamp should be used $=c\sqrt{B\div E}$, where B = cost of lamp per c.-p., E = cost of 1,000 watt-hours of energy, and $c=1{,}410$ when the increase of watts per c.-p. per hour of use $=0.001$ ($c=1{,}000$ when increase $=0.002$, and 815 when increase $=0.003$).

The Tantalum Incandescent Lamp has a fine wire of this rare metal in place of the ordinary carbon filament. Properties of tantalum: melting point = 2,300° C., sp. heat = 0.0365; sp. g. = 16.5; sp. resistance (1m. × 1mm.^{2}) = 0.165 ohm. The resistivity increases with the temperature and at 1.5 watts per c.-p. = 0.855. Lamps (1.5 watts per c.-p.) have a useful life of 400 to 600 hours.

Illumination. Arc lamps: for outdoor or street illumination, 100 to 150 sq. ft. per watt; for railway stations, 10 to 18 sq. ft. per watt; for large halls, exhibitions, etc, 2 sq. ft. per watt; for reading-rooms, 1 sq. ft. per watt and for intense illumination 0.5 sq. ft. per watt.

Incandescent lamps: (16 c.-p.). Ordinary illumination, sheds, depots, etc., 1 lamp (8 ft. from floor) for 100 sq. ft.; waiting-rooms, 1 lamp for 75 sq. ft.; stores and offices, 1 lamp for 60 sq. ft. Dark walls require an increase in the above figures. Nernst lamps, having a "glower" formed of metallic oxides which becomes incandescent during the passage of current, are made in sizes from 25 to 150 c.-p. and require about 1.6 watts per c.-p.

ELECTRIC TRACTION.

Tractive Force and Power. The force, F, required to bring a car from rest to a certain speed, s, (in miles per hour,) within a given time, t, (in seconds,) is F (in lbs.) $=f+\frac{91.1Ws}{t}+20Wp$, where W = weight of car in tons, $f=(20 \text{ to } 30)\times W$, and p = per cent of grade.

It takes a pull of about 70 lbs. per ton to start a car on a level or to round a curve. If there is a grade, the starting pull in lbs. $=(70+20p)W$, based on a speed of 9 miles per hour being attained in 20 secs.

The average H.P. required $=0.00133Fs\div\eta$, where η = efficiency of motor (from 50 to 60%). The per cent grade, p, at which slipping occurs when car is starting $=\frac{100a}{x}-3.5$, where a = ratio of adhesive force to weight on drivers, = 0.125 to 0.16, and x = weight on drivers ÷ total weight of car. When running, $p=\frac{100a}{x}-1.5$.

Resistance of Rails used for Returns. Cir. mils of cross-section of a rail $=124{,}750W$; equivalent cir. mils of rail section in copper $=20{,}800W$; Resistance of a single rail per mile in ohms $=2.5\div W$, approx. (Varies from 2.5 to 5 according to the chemical composition of rail.) W = weight of rail in lbs. per yard.

Safe Current for Feeders, in amperes, $=\sqrt{(\text{diam. in mils})^3\div 1{,}300}$.

Heavy Electric Railroading. Train resistance, R, in lbs. per ton of 2,000 lbs. $=3+1.67s+0.0025\frac{As^2}{w}$, where s = speed in miles per hour, A = cross-section of car in sq. ft., w = weight of train in tons of 2,000 lbs. This formula was found applicable to conditions met with on the Long Island Ry. (W. N. Smith, A. I. E. E., 11–25, 1904).

A formula due to Aspinall is said to give satisfactory results: R (in lbs. per metric ton of 2,200 lbs.) $=2.5+s^{\frac{5}{3}}\div(51+0.028L)$, where L = length of train in feet. The starting resistance varies according to the wheel diameter, condition of track, etc. Aspinall gives as a fair average 17 lbs. per ton of 2,200 lbs. for best conditions.

Electric Passenger Locomotive (N.Y.C. & H.R.Ry.). Type 2-8-2: drivers 44 in. diam.; trucks, 36 in. diam.; diam. of driving axles = 8.5 in.; wheel-base of drivers = 15 ft., total wheel-base = 27 ft. Weight on drivers = 138,000 lbs.; on trucks, 52,000 lbs.; total weight = 190,000 lbs.

Power: direct current, 600 volts; 4 motors, each 550 rated H.P. Maximum power = 3,000 H.P. Normal full-load current = 3,050 amperes. Max. current = 4,300 amp. Normal draw-bar pull = 20,400 lbs., max pull = 32,000 lbs. Speed with a 500-ton train = 60 miles per hour. (General Electric Co., builders.)

ADDENDA.

Large Gas Engines. Belgian and German Practice. Compression, 170 to 200 lbs. per sq. in.; m.e.p. generally taken as 70 lbs. per sq. in. Cooling-water per B.H.P. per hour: cylinders, cylinder-ends and stuffing-boxes, 4 to 5.25 gal.; pistons and piston-rods (hollow), 1.75 to 2.75 gal.; valve-boxes, seats and exhaust-valves, 0.88 to 1.38 gal. (Water entering at 60° F. and leaving at 95° to 115° F.) Engines are started by compressed air (150 to 250 lbs. per sq. in.) and the lubrication is effected by means of a forced oil-feed. The foregoing for engines of 200 to 1,000 H.P.

An Otto-Deutz 4-cycle, double-acting engine (223 B.H.P.) using suction-producer gas made from Belgian anthracite (14,650 B.T.U. per lb.) required 0.704 lb. of dry coal per B.H.P. hour. (R. E. Mathot, Liege Meeting of I. M. E., 1905.)

Shearing Strength of Rivets in lbs. per sq. in. Single-shear: Iron, 40,000; steel, 49,000. Double-shear: Iron, 78,000; steel, 84,000. Distance from center of rivet hole to edge of plate should be about $2d$. (E. S. Fitzsimmons, Master Steam-Boiler Makers' Convention, 1905.)

A safety factor of $4\frac{1}{2}$ should be employed. In butt-joints with two butt-straps or cover plates the rivets are in double shear (page 21).

Flow of Air in Metal Pipes. $Q=c\sqrt{\frac{Fd^5}{L}}$, where d = side or diam. in in., F = friction in ounces per sq. in., L = length in ft., Q = cu. ft. per min., $c=4.4$ for round and 5.5 for square pipes. For a 90° bend in the pipe, add E feet to L. ($E=kd$.) Let r = mean radius of bend in in. Then, when

$r\div d=$	0.5	1	1.5	2
$k=$	5	4	3	2

(J. H. Kinealy, E. N., Aug. 10, 1905.)

When $r=2.5\,d$ the bend offers the least resistance, and E (in inches) = 3.38 × length of the curved portion of pipe, measured along the center line at radius r. (C. W. L. Alexander, Trans. I. C. E., 1905.)

APPENDIX.

MATHEMATICS.

Metric H.P. (Force de cheval). 1 metric H.P. = 75 m.-kgs. per sec. = 542.475 ft.-lbs. per sec. = 0.9863 British H.P. (1 British H.P. = 1.01389 metric H.P.). 1 meter-kilogram (m.-kg.) = 7.233 ft.-lbs. 1 ft.-lb = 0.138255 m.-kg.

Guldinus' Theorems for Areas and Volumes.

1. If a straight or curved line in a plane revolves about an axis lying in that plane, the area of the surface so generated is equal to the length of the line multiplied by the distance through which its center of gravity moves.

2. If a plane area revolves about an external axis in the same plane, the volume of the solid so generated is equal to the area of the figure multiplied by the distance through which its center of gravity moves.

Centers of Gravity of Lines. Straight line: Its middle point. Circumference of a triangle: Form an inner triangle by connecting middle points of sides and inscribe a circle; the center of circle is c. of g. desired. Circumference of parallelogram: At intersection of diagonals. Circular arc: On middle radius at distance x from center of circle [x = (chord × radius) ÷ length of arc]. For very flat arcs c. of g. lies $\frac{2}{3}h$ from chord, where h = height of arc.

MATERIALS.

Metals, Properties of.

	Sp. G.	Lbs. per cu. in	Fusing-points.
Antimony	6.7	0.242	806° F.
Bismuth	9.8	.354	516
Lead	11.38	.411	620
Manganese	8.	.289	3,452
Nickel	9.	.325	2,678
Platinum	21.5	776	3,272

Alloys. **Sterro Metal** (Tensile strength T.S. = 60,000 lbs. per sq. in.): 55% Cu + 42.36% Zn + 1.77% Fe + 0.1% Sn + 0.83% P. **Wolframinium:** 0.375% Cu + 0.105% Sn + 98.04% Al + 1.442% Sb + 0.038 W. **Magnalium:** 2 to 25% Mg + 98 to 75% Al. Sp. g., 2.4 to 2.54; fusing-point, 1,100° to 1,300° F. With 10% Mg, alloy has properties of rolled zinc; with 25%, those of bronze. **Parsons' Manganese Bronze:** 60% Cu + 37.5% Zn + 1.5% Fe + 0.75% Sn + 0.01% Mn + 0.01% Pb (for sheets); 56% Cu + 42.4% Zn + 1.25% Fe + 0.75% Sn + 0.5% Al + 0.12% Mn (for sand castings). T.S. = 70,000 lbs. per sq. in.; elastic limit, 30,000 lbs.; elongation in 6 in. = 18%; reduction of area = 26%.

Nickel-Vanadium Steel. (Carbon content = 0.2%.) With 2% Ni, and 0.7% V, tensile strength = 90,000 lbs. per sq. in; increasing V to 1%, T.S. = 120,000. With 12% Ni and 0.7% V, T.S. = 200,000; increasing V

to 1%, T.S. = 220,000. By tempering the 90,000 lb. steel (heating to 1,560° F. and quenching in water at 68° F.) its T.S. is raised to 168,000. Elastic limits about 80% of T.S. Elongation for 2% Ni steels about 22%; for 12% Ni = 6%.

Malleable Iron, Ultimate Strength. Round bars tensile strength = 43,000 lbs. per sq. in., approx.; elongation = 7% in 8 in.; reduction of area = 3.75%. Square and star-shaped sections have about 85% of the strength of circular sections. Compressive strength is from 31,000 to 34,000 lbs. per sq. in. (Mason and Day.)

Steel. Each per cent. of the carbon content of a steel is divided into 100 parts, each of which is called a "point"; thus, a 40-point carbon steel is one containing 0.4% of carbon.

Portland Cement Concrete in Compression (safe strength). f_c (direct compression) $= 4{,}260 \div (s+g+4.4)$, where s and g are the No. of parts of sand and gravel in the mixture to one part of cement (c). For one cubic yard of concrete, No of bbls. of cement, $N = 11 \div (c+s+g)$; No. cu. yds. sand $= 0.141Ns$; No. cu. yds. gravel or crushed stone $= 0.141Ng$. (1 bbl. = 3.8 cu. ft.)

STRENGTH OF MATERIALS.

Elastic Limit. Yield-Point. Permanent Set. The elastic limit is the point at which the strains begin to increase more rapidly than the stresses causing them. This increase of strain is initially slight but becomes marked later at what is called the "yield-point" (e.g., when scale-beam of a testing machine suddenly drops). That part of the strain which does not disappear when the stress is removed is called the "permanent set." If none of the strain disappears on removal of the stress, the material is said to be "plastic," if the greater part remains, the material is "ductile," and if the material breaks under very low stress and slight stretch, it is said to be "brittle."

Transverse Elasticity (see page 18). In formula $C = f_s \div \partial_s$, ∂_s is the strain between two shear planes 1 in. apart.

Pure Shear Stress (Ultimate) $= C \times$ ult. tensile stress, where $C =$ 1.2 (1.1 to 1.5) for C. I., 1.25 for phosphor bronze and yellow brass, 0.9 for gun-metal, 0.6 for alloy bronzes, 0.75 for W. I., and 0.12 carbon steel, and 0.65 for 0.70 carbon steel. (E. G. Izod, *Engineer*, London, 12–29–'05.)

Aluminum (99% pure). Breaking and safe stresses in lbs. per sq. in.:

	Tension.		Compression.	
	Breaking.	Safe.	Breaking.	Safe.
Castings	14,000–18,000	3,500–4,500	16,000	3,000
Sheets, bars	25,000–40,000	6,000–7,000	20,000	5,000
Wire	30,000–35,000	....	(E = 11,500,000 for cast metal.)	

Allowable Fiber Stresses in Lbs. per Sq. In. (Bach

	W. I.	Steel.			C. I.
		Low Carbon.	High Carbon.	Castings.	
Tension, f_t =	12,800	12,800 17,000	17,000 21,300	8,500 12,800	4,300
Compression, f_c =	12,800	"	"	12,800 17,000	12,800
Bending, f_b =	"	"	"	10,700 15,000	(a)
Shearing, f_s =	10,200	10,200 13,700	13,700 17,000	6,800 12,000	4,300
Torsion, f_{tw} =	5,100	8,500 12,000	12,800 17 000	"	(b)

(The higher values are for homogeneous metal, not too soft.)
(a) For rect. sections, 7,300; circular, 8,800; I sections, 6,200.
(b) For circ. sections, solid and hollow, 4,300; elliptic and hollow rect.,

4,300 to 5,300; rect., square, I, channel, angle, and cruciform sections, 6,000 to 8,000.

The values above given are for constant stresses due to a dead load, P.

For repeated stresses:

(1) load fluctuating between 0 and $+P$, take $\frac{2}{3}$ of tabular values;
(2) " " " $+P$ " $-P$, " $\frac{1}{3}$ " " "

For spring steel (1), $f_b = 52{,}000$ (unhardened) or 62,000 (hardened).

Strength of Cylinders. According to Prof. C. H. Benjamin, if the flanges of a C. I. cylinder are unsupported, the initial fracture will be circumferential, near the flanges, and will be caused by a pressure much less than $p = 2ft \div d$. Also, if flanges are sufficiently braced by brackets to insure longitudinal fracture, a considerable allowance (say $\frac{1}{3}$) must be made for bending and other accidental stresses. Hydraulic cylinders under pressures above 3,000 lbs. should be made from air-furnace iron or steel castings, as water will ooze through ordinary, open-grain C. I. walls <4. in. thick. (A. Falkenau, *Am. Mach.*, 1–4–'06.)

The thickness, t, of the walls of a cylinder under internal pressure, p, may be found from the following formula, which is a simplification by the author of a rather unwieldy one due to C. Bach: $t = 0.42pd \div (f_t - p)$, where d = diam. of cylinder and f_t = allowable stress in the material employed (to be used only when $p < 0.77f_t$).

Values of f_t: C. I. and bronze, 4,300 to 8,500 (and even 10,000 for strong iron); phosphor-bronze, 7,100 to 14,200; cast steel, 14,200 to 17,000 (for Mannesmann tubes of Martin steel, 18,000 to 43,000); W. I., 12,800 to 25,600. t and d in in., p and f_t in lbs. per sq. in.

Cotter Joints (W. I. and Machinery Steel). Diam. of rod, d, is enlarged to $D(=1.33d)$ in socket. Socket diam. $= 2D = 2.66d$; thickness of key (steel) $= 0.25D$; mid-depth of key, $h = 1.33D = 1.75d$. Ends of socket and rod should extend $\frac{2}{3}h$ to $\frac{3}{4}h$ beyond key slots ($=1.25d$, average).

Fly-Wheels, Safe Velocities for. Velocity in ft. per sec. $= 1.63\sqrt{s\eta f_t \div w}$, where s = factor of safety, η = efficiency of joint used, w = wt. of 1 cu. in. of material, and f_t = tensile stress of material.

		Hard Maple.	Cast Iron.	Steel.
w	=	.0283	.261	.283
f_t	=	10,500	10,000	60,000
s	=	40	10	20

In wooden rims $s = 20$, but as the segments break joints in assembling the strength is reduced one-half, making s really equal to 40. Steel rims are made up from segments riveted together, and the usual factor 10 is similarly increased to 20. Using above values and considering wheels as solid, $\eta = 1$. For cast-iron rims, $\eta = 0.25$ for flange-joints between arms, $= 0.5$ for pad-joints (each arm having a flat enlarged face on its end to which rim-sections are bolted), $= 0.6$ in heavy, thick-rimmed balance-wheels with joints reinforced by steel links which are shrunk on. (W. H. Boehm, in *Insurance Engineering*.)

Riveted Joints. General Formulas. (W. M. Barnard.)

$$d = \frac{4t}{\pi}\left(\frac{nf_c + mf_c'}{nf_s + 2mf_s'}\right); \quad p'' = \left(\frac{nf_c + mf_c'}{f_t}\right)d + d, \; = \frac{\pi d^2}{4}\left(\frac{nf_s + 2mf_s'}{tf_t}\right) + d.$$

Efficiency of joint $= 1 \div [1 + f_t \div (nf_c + mf_c')]$.

In the above n = No. of rivets in single shear in a unit strip equal to the max. pitch (where rows have different pitches), and m = No. of rivets similarly in double shear. f and f' are respectively strengths in single and double shear.

f_t (iron) varies from 40,000 for single-riveting, punched holes to 50,000 for double-riveting, drilled holes. f_t (steel) = 55,000 (punched holes) to 60,000 (drilled holes). f_s (iron) = 36,000 to 40,000; f_s (steel) = 45,000 to 48,000.

f_c (iron) = 67,000 (for lap-joint) and 90,000 (for butt-joint); f_c (steel) = 85,000 (lap) and 100,000 (butt).

Helical Springs of Phosphor-Bronze will withstand the action of salt-water. For wire up to $\frac{3}{8}$ in. diam. use formulas on pages 23 and 24, taking $f_s = 17{,}825$, and $C = 6{,}200{,}000$. (H. R. Gilson, *Am. Mach.*, 7–19–'06.)

Moment of Inertia. The following graphic method is in extended use among designers of structural steel.

Divide area of section A (Fig. 38) into 10 or more strips parallel to direction of neutral axis desired, and set off lengths representing their respective areas on the polar diagram at the left, as 01, 12, 23, . . mn. These strip areas are to be considered as parallel forces which act at their respective centers of gravity as indicated by the small circles. Set off pole O, making $OB = \frac{1}{2}A$, and draw $O0$, $O1$, $O2$, . . . On. Draw $K0 \parallel O0$, $01 \parallel O1$, $12 \parallel O2$ . . . , closing diagram with $nL \parallel On$. At J, the intersection of nL and $K0$, draw JX, which is the neutral axis of the section. Find the

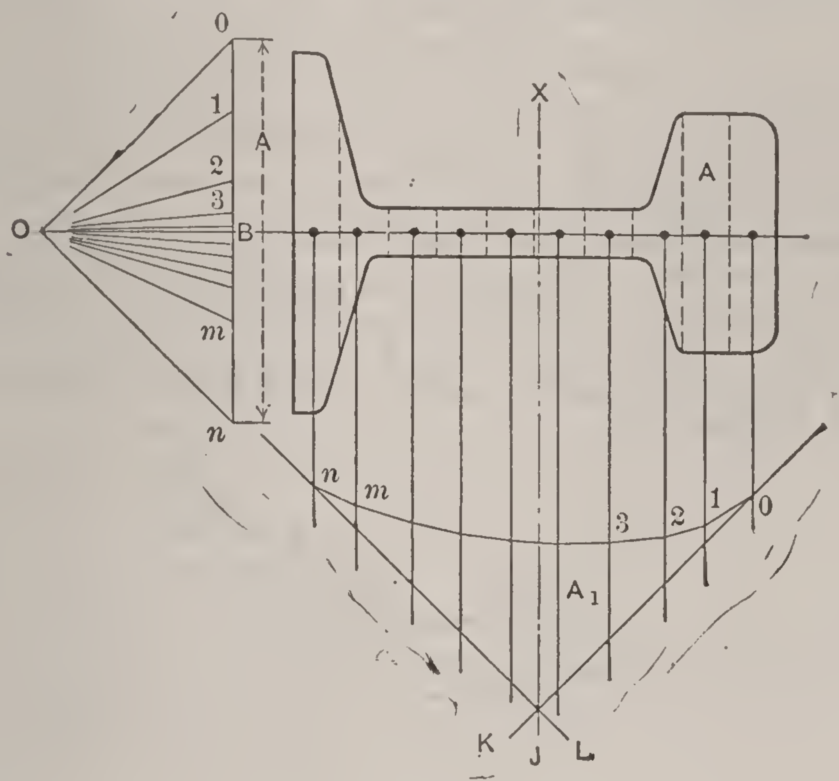


FIG. 38.

area of the equilibrium polygon, A_1, then, Moment of Inertia of Section $=$ area $A \times$ area A_1. (The greater the number of strips, the more accurate the results obtained.)

Laminated Springs. For nearly flat springs, Deflection $\Delta = Wl^3 \div 4{,}460nbt^3$ (approx.), but for exact results, as true for buffing as for ordinary springs, Deflection $= \Delta[1 - c(5c - 7\Delta) + 3\Delta^2] \div 3l^2$, where l = length of arc of top plate, c = camber, b and t = width and thickness (all in inches), n = No. of plates, and W = load in tons. (H. E. Wimperis, *Engineer*, London, 9-15-'05.)

Strength of Forged Rings (for hoisting, etc.). Consider the suspended ring to be divided into two equal parts by a vert. plane, $\frac{1}{2}$ of total load W_1 acting on each half Employ formula for combined tension and bending (page 29) $\cdot$ $f_t = W\left(\frac{1}{a} + \frac{r}{cs}\right)$, where $W = \frac{W_1}{2}$, $a = \pi d^2 \div 4$, $r = 0.5(D + d)$, where D = internal diam. of ring and d = diam. of iron used, $c = 1.6$ for W. I. or steel, $s = \pi d^3 \div 32$. This reduces to: $f_t d^3 - 2.23 W_1 d = 1.6 W_1 D$, in which $f_t = 5{,}000$ to $6{,}000$ lbs. per sq. in. for safe tensile stress (allowing for suddenly applied load and efficiency of weld). W_1 in lbs., d and D in in., any two of which being assumed, the third may be derived from formula.

A formula discussed in *Engineering* (London), 5-29-'95, and arrived at through a different method, is: $f_t d^3 - 1.62 W_1 d = 1.62 W_1 D$.

Columns. Euler's Formulas. Safe load $W = c\pi^2 EI \div sl^2$, where $c =$ 0.25 for one fixed and one free end, $= 1$ for both ends free, load guided, $= 2$

for one fixed end and one free end, with load guided, $=4$ for both ends fixed, load guided; s = safety factor = 5 to 6 for W. I. and steel, 8 or more for C. I., and 10 for fir. The above formula should not be used where l (= length in in.) is less than $25d$ for W. I. and steel, or less than $12d$ for C. I. and wood, where d = diameter or smaller rectangular dimension of cross-section in in.

For reinforced-concrete columns, $c=1$, $s=10$, and $E=(a+b)E_c \div (a+1)$, where E_c = modulus for concrete, a = concrete cross-section ÷ steel cross-section, $b=E_{steel} \div E_c$.

For shorter bars subjected to thrust, the following formula, due to Grashof, should be employed:

$$W = \text{max. load in lbs.} = ckaI \div \left(\frac{al^2}{C} + cI\right),$$

where a = sectional area of bar in sq. in.; k = 12,000 for steel (= 10,000 for W. I.); C = 5,000 for steel and 5,600 for W. I.; $c=1$ for bar free at both ends (e.g., connecting-rod), $=4$ for bar fixed at both ends. For connecting-rods take but 75% of the above values for k.

Collapse of Tubes. (Lap-welded Bessemer steel, 3 to 10 in. in diam.) Collapsing pressure p, in lbs. per sq. in. $=1{,}000(1-\sqrt{1-(1{,}600t^2 \div d^2)})$, where $(t \div d) < 0.023$; $p=(86{,}670t \div d)-1{,}386$, where $(t \div d) > 0.023$. (Approx., $p=50{,}210{,}000(t \div d)^3$ when $(t \div d) < 0.023$.) These formulas apply when $l \geqq 6d$. A safety factor of from 3 to 6 should be introduced, its size being according to the risk at stake to life and property. (R. T. Stewart, A. S. M. E., May, '06.)

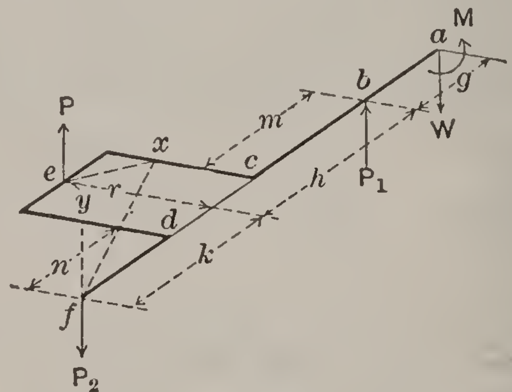


FIG. 39.

Torsion and Bending (see also page 31). According to Bach, Equivalent Bending Moment $=0.35B_m+0.65\sqrt{B_m{}^2+(\alpha T_m)^2}$, where $\alpha=1.9$ for W. I., 1.15 for soft steel, and 1 for hard steel.

Cranked Shafts. Let *abcedf* (Fig. 39) be a horizontal cranked shaft. The turning force P (having a moment M, due to wt. of fly-wheel at a and equal to Pr) acts at center (e) of crank-pin in the direction indicated. Weight of fly-wheel (W) acts vertically downward at a. Neglecting end thrust:

Bearing reaction at center b (upward) $\quad = P_1 = \dfrac{W(g+h+k)}{h+k} - \dfrac{Pk}{h+k}$;

" " " " f (downward) $= P_2 = \dfrac{Wa}{h+k} + \dfrac{Ph}{h+k}$.

Bending and Twisting Moments: at b, $B_m=Wg$, $T_m=Pr$; at c, $B_m=W(m+g)-P_1m$, $T_m=Pr$; at d, $B_m=P_2n$, $T_m=0$; at e, $B_m=P_2k$, $T_m=P_2r$; at x (any point on throw) the moments $P \cdot ex$ and $P_2 \cdot fx$ are each to be resolved into moments in and also perpendicular to the cross-section and then combined. The component in the plane of cross-section gives T_m and

the component perp. to cross-section gives Bm. Similarly for any point y. P should also be assumed as acting downward and above values worked out for that direction.

Gap Frames for Riveters, Punches, Shears, etc.

The size and character of the work determine the depth of throat l, or distance from point of application of force w to the nearer or tension flange of frame. Assume the main section of frame (lying in a plane ⊥ to direction of force w) to be of an I-, or equivalent box-section, of area a, and having a uniform tensile stress $\frac{w}{a}$ (due to w) distributed over it. Determine position of the neutral axis of section and also its moment of inertia, I.

The bending moment Bm (due to w) on the section $= wl_1 = w(l+x)$, where x = distance from neut. axis to outer fibers of the tension flange. Tensile stress due to $Bm = Bmx \div I$, and total stress in tension flange $= \frac{Bmx}{I} + \frac{w}{a}$ (1).

Similarly, stress in compression flange due to $Bm = \frac{Bmy}{I}$, where y = dist. from neut. axis to outer fibers of compression flange.

This stress is opposed by the uniformly distributed tensile stress, $\frac{w}{a}$, and the net stress in compression flange $= \frac{Bmy}{I} - \frac{w}{a}$ (2).

If (1) and (2) differ from the safe stresses for the material employed (C. I., or cast steel) the area and proportions of section must be altered until substantial agreement is arrived at. Sections parallel to direction of force w are calculated for bending only, there being no direct stress $\left(\frac{w}{a}\right)$ on them, but the webs must have sufficient surplus section to resist shear.

Steel Chimneys (self-supporting). H = height; D and D_1 = outside and inside diams.; T = thickness $[=0.5\,(D-D_1)]$; Db = diam. of bell-shaped base ($=1.5D$ to $2D$); Hb = height of base ($=Db$). All dimensions in feet. Wind pressure P (lbs. per sq. ft.) = (velocity in miles per hour)$^2 \div 200$. P is generally taken at 50 lbs., or 25 lbs. *actual* pressure per sq. ft. of projected area (HD). To this is added 5 lbs. to allow for compression on one side, making $P_{gross} = 30$. Bending moment, $Bm = 30HD \times 0.5H = 15DH^2$. Section modulus, $S = \frac{\pi}{32}\left(\frac{D^4 - D_1{}^4}{D}\right) = 0.7854D^2T$. F (per sq. ft.) $= Bm \div S = 19.1H^2 \div DT$, or f (sq. in.) $= 0.1326H^2 \div DT$. For steel plates $f = 45{,}000$ to $50{,}000$, or taking strength of riveted joint as 36,000 and safety factor of 4. $f_{safe} \leq 9{,}000$. To find T at any section, measure H from top of chimney to section and substitute in formula.

Total wind pressure $P_1 = 25HD$ lbs., or, if H and D are expressed in inches, $P_1 = 0.1736hd$ lbs. Resistance to breaking at foundation $= 1.57db^2tf \div h$, where db, t, and h are inch equivalents of Db, T, and H. For stability, make Df (of foundation) $= Hf = \frac{H^2D}{26{,}000} + 10$. Moment of wind pressure $= P_1(0.5H + Hf)$. Let W = total weight of chimney, lining, and foundation, in lbs.; then, x, or the lever-arm of W, $= P_1(0.5H + Hf) \div W$. If $x < 0.5Df$, the structure will be stable. ($0.5Df \div x$ = factor of stability, usually about 1.6, but increased to 2.5 and even 3 for loose soil.) t should never be taken less than $\frac{1}{8}$ to $\frac{3}{16}$ in., to insure durability, rivet diam., dr, not less than $\frac{1}{2}$ in., spaced about $2.5dr$ (c. to c.), and in any case $< 16t$. (1 cu. ft. of foundation weighs 125 to 150 lbs.)

Foundation bolts (usually 6 or 12): Gross overturning moment $= 12.5DbH^2$; moment resisting overturning $= 0.5W_1Db$ (where W_1 = wt. of shell), and net overturning moment $T = 0.5Db(25H^2 - W_1)$. If D_c = diam. of bolt circle, then Tc (or overturning moment at Dc) $= 0.5Dc \times 9{,}000$ lbs. $\times$ No. of bolts $\times$ area of 1 bolt in sq. in. ($T_c = D_cT \div Db$).

Lining: Where temperatures are above 600° F., fire-brick linings are used. Linings are generally 9 in. thick for lower 30 ft. of stack, and 4 in. thick above that height. 1 cu. ft. brick (red or fire) weighs about 120 lbs

ENERGY AND THE TRANSMISSION OF POWER.

Screws for Power Transmission (Screw-Presses, etc.). Square threads are preferable to V threads, and the moment to raise load W

$$=Wr\left(\frac{p''+2\pi r\mu}{2\pi r-p''\mu}\right),$$

where r = mean radius of thread, p'' = pitch, and μ = coeff. of friction between nut and screw. Let n = No. of threads in nut, the projected area of which $=0.7854n(d_1^2-d_2^2)$, and $W=0.7854np(d_1^2-d_2^2)$, where d_1 and d_2 are root and outer diams. of thread, and p = allowable pressure in lbs. per sq. in. of projected area, $=125{,}000 \div V$, where V = rubbing speed in ft. per min. and $\leqq 100$. ($p=80{,}000 \div V$ when $V=400$.) These values of p are for W. I.; for steel, multiply same by 1.2. $\mu=0.07$ for heavy machine-oil and graphite in equal vols., $=0.11$ for lard-oil, $=0.14$ for heavy machine-oil.

Efficiency: Let α = pitch angle at radius r, ($\tan \alpha = p'' \div 2\pi r$), and ϕ = angle of friction, ($\tan \phi = \mu$). Then, efficiency $=\tan \alpha \div \tan(\alpha+\phi)$. For max. eff., make $\alpha=45°-0.5\phi$. In order that load may not overhaul, α must be less than ϕ, and the efficiency cannot then exceed 50%.

Piston-Rods, Connecting-Rods, Eccentric-Rods.

Euler's formula for compression (both ends free) is : $P=\pi^2EI \div l^2$, where P = total pressure or load in lbs., l = length of rod in in. ($I=\pi d^4 \div 64$ for circular sections; $I=b^3h \div 12$ for rectangular sections).

Substituting in formula, introducing a factor of safety s, and taking E = 29,000,000 for W. I. forgings, $P=29{,}000{,}000\ d^4 \div 2sl^2$ for circular sections, and $P=23{,}800{,}000b^3h \div sl^2$, where d = diam. in in., b and h = breadth and height in in.,—d, b, and h being taken at mid-length. For piston-rods, $s=8$ to 11 when load fluctuates between P and 0; $s=15$ to 22 when load fluctuates between $+P$ and $-P$. (For very large horizontal engines the deflection of rod due to weight of rod and piston should be considered, and it should not exceed 0.15 in.) For eccentric-rods $s=40$, for connecting-rods $s=25$ and 15 respectively for circular and rectangular sections. h at mid-length $=1.75b$ to $2b$ (heights at crank and cyl. ends $=1.2h$ and $0.8h$, resp.). d tapers to $0.8d$ at crank end, and to $0.7d$ to $0.75d$ at cyl. end). For very low speeds (circ. section) $s=33$; for sudden changes in direction of P (as in pumps), $s=40$ to 60. For high speeds, as in locomotives (rect sections), $h=2b$ $s=6.6$ to 3 3 (See also Columns, Euler's formulas, *ante*.)

Connecting-Rod Ends (Marine type, rod formed with a T-end, brasses being held to T by bolts and cap). Diam. of each bolt at bottom of thread, $d=0\,02\sqrt{P}$, where $P=\frac{1}{2}$ max pressure on piston in lbs. Thickness of cap and T on end of rod, $t=1.4d$. These values of d and t are for W. I.; for steel take 90% of same.

Piston-Rings. Radial depth, $h=0.033d$ when bored concentrically, $=0.04d$ opposite joint when bored eccentrically (tapering to $0.7h$ at ends). Width $=2h$; overlap of ends $=0.1d$, where d = diam. of cyl.

Stuffing-Boxes. Inner diam. of box = depth $=d+(0.8$ to $1)\sqrt{d}$, where d = diam. of rod

Pedestals (d = shaft diam., l = length of brass, both in in.). Diam. of bolts for base and cap $=0.25d+0.125$ in.; dist. bet. centers of cap-bolts $=3.3d+1.65$ in.; do., base-bolts $=3.5d+1.75$ in.; width of pedestal $=0.72l$. Thicknesses : cap, $0.375d$; base-plate, $0.25d+0.125$ in.; metal around cap-bolts and brasses, $1.8d+0.09$ in. (If $d<7$ in., use 4 bolts each for base and cap.) Brasses: thickness at center $=0.08d+0.125$ in.; do., at sides, $0.06d+0.09$ in.

Journal Bearings.

Allowable pressures (p) per sq. in. of projected area ($l \times d$):

Journal.	Bearing.	p.
Crucible steel (hardened)	Crucible steel	2,100 lbs.
" " (hardened)	Bronze	1,250
" " (soft)	"	850

Journal.	Bearing.	p.
Wrought iron, polished	Bronze	570
W. I. or C. I.	"	425
" " "	C. I. or bronze	350
W. I. (water lubrication)	Lignum Vitæ	350

	Speed.	
Crank-pin	Moderate	900
"	High	550
Cross-head pin	Moderate	1,100
" "	High	700
Main bearings,		200–350
Crank- and cross-head pins, locomotives,		1,400, 2,100 resp.
Crank-pin for punch and shears,		2,800 and up (Bach.)
Main rods of locomotives,		1,800–2,100
Freight-car axles,	$l=1.8d$	350
Passenger-car axles,	$l=1.8d$	300
Neck bearings of sheet-mill rolls,		1,000–1,500

(Eng'rs Soc. of W. Pa., Dec. '05.)

Main bearings of engines, $c \div \sqrt{v}$

[v=vel. of rubbing surface in ft. per sec.; $c=500$ for vertical engines, $=375$ for horizontal. (Edwin Reynolds.)]

$pV<50\,000$; $p=30$ to 80 lbs.; $V=400$ to 1,200 ft. per min.; $l=3d$. Allowance in diam. for oil-film$=0.001(d+1)$ in. for $d\leq 5$ in. Allowance$=0.001(d+4)$ in. for $d>5$ in. (Gen. Elec. Co. Practice.)

Thrust-bearings: $pV=40{,}000$ to 50,000, with loads up to 1,000 lbs. per sq. in. of projected collar area.

Worm-gears: $pV=60{,}000$ to 75,000 for max. efficiency, the higher value for high values of V, and where helix angle$=20°$; worm of hardened steel, wheel of phosphor-bronze. For electric-elevator work $V=600$ to 1,000 ft. per min.

Large shaft-bearings tested by the Westinghouse E. & M. Co., over runs of 7 hours yielded the following unusually high values for pV: 9-in. shaft, 150,000 to 500,000; 15-in. shaft, 260,000 to 840,000 ($p=140$ to 170). Lower values for each size were when heavy machine oil was used, higher values with paraffin oil. (A. S. M. E., Dec. '05.)

Friction Couplings (C. I.). Shaft diam.$=d$; hub diam.$=2d$; depth of groove$=\frac{1}{8}d$; width of groove$=\frac{1}{2}d$; width of friction-cone faces$=1\frac{1}{8}d$; thickness of wheel webs$=\frac{3}{8}d$; angle between shaft and cone faces$=4°$ to 10°.

Claw Couplings (C. I.). Diam. of both claws, $D=2.1d+2$ in.; diam. of fixed hub$=1.6d+1.6$ in.; length of fixed claw$=0.9d+1$ in.; depth of recesses in both claws$=0.6d+0.6$ in.; length of fixed hub$=0.5d+0.5$ in.; length of sliding claw$=1.7d+1.7$ in. (of diam. D throughout length); depth of groove midway between end and recess$=0.3d+0.3$ in.; width, do.,$=0.5d+0.5$ in. (d=diam. of shaft).

Roller Bearings. For heavily loaded, slow-running journals, $P=2{,}100nld$ for hardened-steel rollers (Ing. Taschenbuch).

The coefficient of friction for roller bearings is from 0.2 to 0.33 of that of plain bearings. (C. H. Benjamin, Machinery, Oct. '05.)

Mossberg bearings (rollers confined rigidly as possible in a cage): Safe load in lbs.$=cnld$, where $c=250$ for rollers up to $\frac{3}{8}$ in. diam. ($c=300$ to 350 for larger rollers). l (generally)$=1.5\times$shaft diam. D. For D up to 12 in., diam of roller $d=0.104D$; above 12 in., $d=1\frac{3}{8}$ to $1\frac{1}{2}$ in. n (approx.)$=27-(1.6\div d)$ for $d<1\frac{1}{4}$ in.; $n=90-(80\div d)$ when $d>1\frac{1}{4}$ in. Take nearest even number.

Ball Bearings. Max. allowable load on one ball in lbs., $P=cd^2$. Values of c: For C. I balls between two planes, $c=35$; steel balls on plane, conical, or cylindrical surfaces, $c=700$ to 1,000; steel balls in races whose radius of curvature$=\frac{2}{3}d$, $c=1{,}400$ to 2,100. Above values for continuous use; for intermittent use c=twice lower values given (d=diam. of ball in in.). Total allowable load on bearing$=0.2P\times$No. of balls. (Bach and Stribeck.)

According to C. Gégauff (*L'Industrie Electrique*, 7–25–'05) the least power is lost in friction when $d=(D\div 7)+0.08$ in., where D=inner diam. of race in in. Max. allowable load in lbs. for an annular bearing, $P=84{,}000D\div(ND+375)$, where N=r.p.m.

For a 2-point bearing, the coeff. of friction, $\mu=0.0015$; for 3-point, 0.003 to 0.006; for 4-point, 0.015 to 0.06 (which is no better than a plain bearing). The friction loss is constant up to linear speeds of 2,000 ft. per min. Above 17,000 r.p.m. centrifugal force causes the balls to slide on the shaft instead of rolling.

Bevel Gearing. θ = angle between shafts $=\alpha+\beta$, where α and β are angles made by the shafts and elements in their respective pitch cones (α for larger gear). Let $\phi=180°-\theta$, and γ = angle to be added to α and β to give face angles of gears Then, if $\theta<90°$, $\tan\beta=r\div[r\cot\theta+(R\div\sin\theta)]$; if $\theta=90°$, $\tan\beta=n\div N$; if $\theta>90°$, $\tan\beta=r\div[(R\div\sin\phi)-r\cot\phi]$. $\alpha=\theta-\beta$; $\tan\gamma=\sin\beta\div0.5n$. Face angles $=\alpha+\gamma$ and $\beta+\gamma$ for larger and smaller gears respectively. $D=D_1+D_2\cos\alpha$; $d=d_1+D_2\cos\beta$. (D, d = outside diams.; D_1, d_1 = pitch diams.; D_2 = working depth of tooth; R, r = pitch radii $(=0.5D_1, 0.5d_1)$; N, n = No. of teeth. Capital letters for larger gear.)

The cutter for larger gear should be the proper one to cut N_1 teeth, where $N_1=N\div\cos\alpha$; for smaller gear, the one to cut n_1 teeth, where $n_1=n\div\cos\beta$.

Spiral Gears. Let angle that teeth make with a line parallel to axis of gear $=\theta$. Then, normal pitch $\tau=p''\cos\theta$ (where p'' = circumferential pitch), and $p''=\tau\div\cos\theta$. Let Pd = diametral pitch, N = No. of teeth in a spur-gear of pitch radius r, and N_1 = No. of teeth in a spiral gear of pitch radius r. Then, $N=2rPd$, and $N_1=2rPd\cos\theta$. Pitch diam. $=N_1\div Pd\cos\theta$; outside diam. = pitch diam. $+(2\div Pd)$.

The teeth of spiral gear should be cut with a spur-gear cutter which is correct for N_2 teeth, where N_2 = (No. of teeth in spiral gear) $\div\cos^3\theta$. γ and γ_1 (page 50) $=(90°-\theta)$ and $(90°-\theta_1)$ respectively.

Worm-Gears. Involute gears of more than 27 teeth, and having addenda of $0.25p''$, yield favorable results for pitches not exceeding 18°. Allowable pressure on teeth, P(in lbs.) $=cbp''$, where b = width of tooth in in., and p'' = pitch in in.

$c=250$ to 400 for cast-iron (=450 to 700 for phosphor-bronze wheel and hardened-steel screw).

Worms whose threads make an angle > 12.5° with a normal to axis of worm generally run well and are durable. (Halsey.)

Diam. of worm wheel at throat $=0.3183\times$(No. of teeth + 2) $\times$ pitch of worm in in.

Power Transmitted by Worm-Gearing. $p^4=(aF^2+bF+c)\div N$, for single thread, where p = pitch of teeth in worm wheel in in., F = H.P. transmitted, and N = r.p.m. For $F>3$ H.P., $a=4.74$, $b=113$, $c=-105$; for $F<3$ H.P., $a=22$, $b=25$, $c=2$.

For double, treble, and quadruple threads take $2N$, $3N$, $4N$, respectively for denominator of formula. Greatest pitch diam. of worm, $d=17.2p\div F$, for single thread. For double, treble, or quad. threads multiply formula value of d for single thread by 2, 3, or 4. The foregoing is for finished worms and gears; if rough, cast teeth are used, multiply values of p and d obtained from formulas by 1.33 and 0.8, respectively. (Derived from practice of Otto Gruson & Co., as stated by W. H. Raeburn, *Am. Mach.*, 4–19–'06.)

Flat-Link Driving Chain (Steel). Load in lbs. $=P$; end diam. of pin, $d=(2.4P+6{,}100)\div(P+27{,}000)$; diam. of pin bet. links $=1.25d$ for small sizes (ranging to $1.12d$ for large sizes); width of link $=2.5d$; length of pin bet. links $=1.65d+0.22$ in. (for $d<1$ in.), or $2.62d-0.7$ in. (for $d>1$ in.); length, c. to c. of pins $=2.7d+0.16$ in.; over-all length of link $=4.4d+0.16$ in. No. of plates, i ($\frac{1}{2}i$ on each side):

When $P=$	up to 1,000 lbs.	1,000 to 4,500	4,500 to 13,000	larger
$i=$	2	4	6	8

Thickness of each plate $=(3.17P+3.900)\div i(P+29{,}000)$.

(Derived from data on a chain extensively used in Germany.)

Pulleys (C. I.). Width of face, $b_1=(1.1\times$belt width$)+0.4$ in.; thickness of rim at edge $=(0.01\times$radius of pulley$)+0.12$ in. Crowning: diam. of pulley at center is $0.12\sqrt{b_1}$ greater than diam. at edges. No. of arms $=0.7\sqrt{d}$. For oval arms h (long axis of ellipse) $=\sqrt[3]{1.25btd\div\text{No. of arms}}$. h_1 (short axis) $=0.4h$. h and h_1 (at hub) taper to $0.8h$ and $0.8h_1$ at rim (b = belt width, t = belt thickness, d = diam. of pulley,—all in in.). Length of hub $=b_1$, when $b_1>1.2$ to $1.5\times$shaft diam. (for narrow faces); for wide

faces, length may be less than b_1. For loose pulleys make length of hub $= 2 \times$ shaft diam. If $b_1 > 12$ in., use two sets of arms.

Pulley Blocks and Sheaves. Diameters are taken considerably less in hoisting work than for power transmission. The Ing. Taschenbuch gives the following: Diam. of sheave $= c \times$ diam. of rope, where $c = 20$ for wire rope and 8 for hemp.

Brakes (Fig. 40). Let W = pressure on brake lever in lbs., P = braking force at rim of wheel in lbs., μ = coeff. of friction $\leqq 0.5$ for wood or leather on iron (dry surfaces) $= 0.18$ to 0.25 for iron on iron, diminishing with increase in vel. For block brakes (I.) $W = \frac{PB}{A+B}\left(\frac{1}{\mu} \pm \frac{C}{B}\right)$, the minus sign being used for rotation indicated,—plus for opposite. For $B \div C = \mu$,

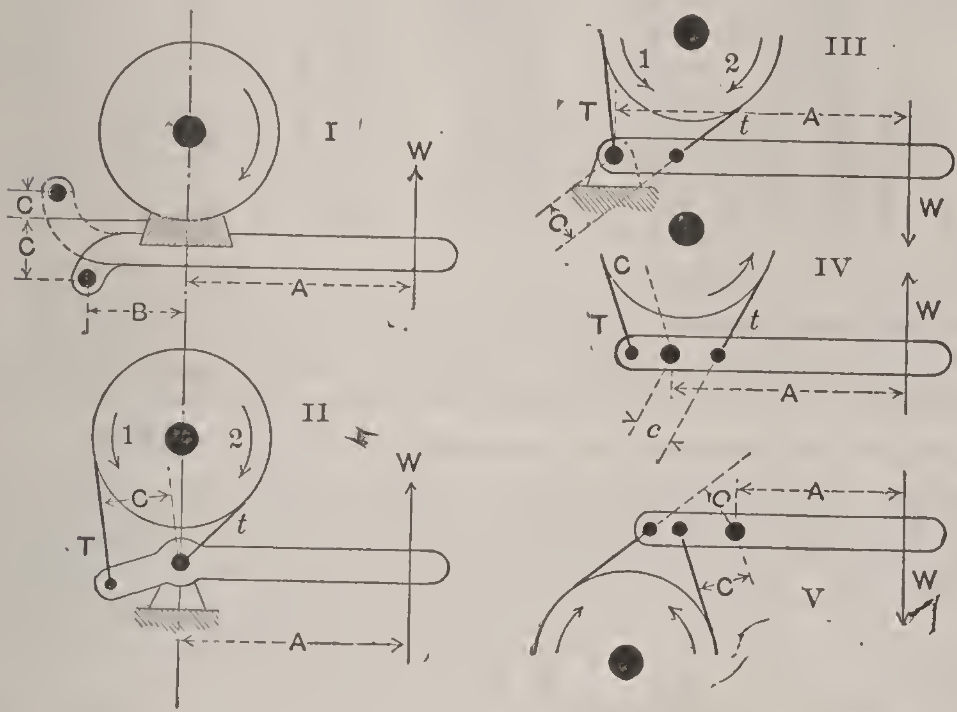


Fig. 40.

$W = 0$, or the brake is self-acting; $B \div C$ is therefore made $> \mu$. For dotted position, C is negative and signs in parenthesis should read $\mp$. For opp. direction of rotation, $B \div C$ should be $< \mu$.

Band Brakes: Let e = base of hyperbolic system of logarithms = 2.71828; α = angle spanned by the arc of contact of band with wheel; t and b = thickness and width of band, and f_t = allowable unit stress in band. Then (in II.) tension $T = P \div (e^{\mu\alpha} - 1)$; and $t = Pe^{\mu\alpha} \div (e^{\mu\alpha} - 1)$, for direction 1 (for direction 2, interchange values of T and t). Band cross-section $= bt = Pe^{\mu\alpha} \div f_t(e^{\mu\alpha} - 1)$, where $f_t = 8{,}500$ to $11{,}000$ lbs. per sq. in. (t is generally about 0 15 in.,—b not more than 3 in.). If μ is taken $= 0.18$ and $\alpha \div 2\pi = 0.7$ (generally), then $T = 0.83P$ and $t = 1.83P$, for direction 1.

For $\alpha \div 2\pi$	$= 0.1$	0.3	0.5	0.7	0.9
$e^{\mu\alpha}$	$= 1.12$	1.40	1.76	2.21	2.77

In II, $W = TC \div A$; in III, $W = tC \div A$. W is least when end with lesser tension is attached to lever, as T in II (direction 1) and t in III (dir. 2).

Differential Brake (IV): $W = (TC - tc) \div A = P(C - ce^{\mu\alpha}) \div A(e^{\mu\alpha} - 1)$. If $C = ce^{\mu\alpha}$, $W = 0$; C is generally taken $> ce^{\mu\alpha}$. (For $\alpha \div 2\pi = 0.7$, $C = 2.5c$ to $3c$.) For alternating directions of rotation (V), $W = PC(e^{\mu\alpha} + 1) \div A(e^{\mu\alpha} - 1)$. A block brake is preferable to this arrangement.

HEAT AND THE STEAM-ENGINE.

Properties of Saturated Steam (below Atmos. Pressure).

p, abs.	$t°$ F.	v.	w.	H.	h.	L.
.089	32.	3,387	.0002952	1091.7	0.	1091.7
.125	40.	2,717	.0003681	1094.2	8.	1086.2
.25	59.5	1,270	.0007874	1100.	27.5	1072.5
.50	80.	640.8	.00158	1106.3	47.8	1058.5
.75	92.5	442.5	.00226	1110.	60.5	1049.5
1.	101.99	334.6	.00299	1113.1	70.	1043.1
2.	126.27	173.6	.00576	1120.5	94.4	1026.1
4.	153.09	90.31	.01107	1128.6	121.4	1007.2
6.	170.14	61.67	.01622	1133.8	138.6	995.2
8.	182.92	47.07	.02125	1137.7	151.5	986.2
10.	193.25	38.16	.02621	1140.9	161.9	979.
12.	201.98	32.14	.03111	1143.6	170.7	972.9
14.	209.57	27.79	.03600	1145.8	178.3	967.5
14.7	212.	26.42	.03794	1146.6	180.9	965.7

Superheated Steam. According to Linde (Z. V. D. I., Oct. 28, '05) the PV law may be expressed as: $144p(v+0.261)=85.86\tau$, where p=lbs. pressure per sq. in., v=cu. ft. in 1 lb. at the pressure p, and τ=absolute temperature in degs. F A formula which expresses the results of his experiments to determine k_p is: $k_p=0.462+p\left(\frac{0.263}{\tau-135}-0.00022\right)$, p and τ as above. Herr Berner (Z. V. D. I., 9-2-'05) states that Linde's values for k_p are confirmed by his own observations, those of Lorenz being from 20 to 25% too high. He further states that the cost of lubrication is slightly higher than when saturated steam is used, that the resistance to flow in a superheater coil$=1.2\times$resistance of smooth pipe, and that the resistance of a valve fully open is equal to the resistance of about 55 ft. of smooth pipe. The velocity of flow in engine passages may be as high as 12,000 ft. per min. (Arndtsen, Z. V. D. I., 11-25-'05.)

Corliss Valves, Dash-Pots. Diam. of valve$=c\times$cyl. diam., where $c=0.25$ for valve on high-pressure cyl. ($=0.2$ for low-pres. cyl.). Dash-pot diameters are about 0.8 of the diams. of their respective valves.

Steam Consumption of Compound Engines, high-grade, at full load =15.6 lbs. per kilowatt-hour (=11.5 lbs. per H.P.-hr.) at 170 lbs. gauge pressure, 90° F. superheat, and 26 in. vacuum. (Averages by Stevens & Hobart, *Power*, Dec. '05.)

Prime Movers for Power Plants. In a high-grade power plant about 10.3% of the heat units in a pound of coal are delivered to the bus-bars in the form of electricity It is possible to raise this thermal efficiency to about 14.4% (with steam-turbines to 15%) by reducing the losses due to the stack, boiler radiation, and leakage, and by using superheated steam. Where the load-factor exceeds 0.25, economizers should be used. Auxiliaries should be steam-driven, with exhaust into heater. The friction loss of a 7,500 H P engine recently tested was 6.35% of the H.P. generated. Large gas-engines can convert about 24% of the energy of coal into electric energy, the chief objection to their use being with regard to overloads. This objection may be overcome by a suggested combination of gas-engines and steam-turbines (utilizing the waste heat of the gas-engines in the production of steam), which would yield an average thermal efficiency of 24.5%.

Comparative cost of maintenance and operation of plants per kw.-hr.:

	Steam-engines.	Steam-turbines.	Gas-engines.	Gas-engines and turbines.
Maintenance and Operation	100	80	51	46.3
Relative Investment	100	83	100	91

Marine Steam-Engines.

The Screw Propeller. The pitch of a screw is the distance which any point in a blade will advance in the direction of the shaft or axis during one revolution, the point being assumed to move around the axis and without "slip." Propellers are generally provided with four blades (naval vessels and small high-speed boats with three). The blades are generally inclined backward from the vertical from 8° to 20° (according to the r.p.m.) in order to throw the water to the rear and to increase the efficiency.

The indicated thrust of screw, $T = (\text{I.H.P.} \times 33{,}000) \div NP$, where N = r.p.m., and P = pitch in feet. The mechanical efficiency of the shaft transmission varies from 0.8 for engines of about 500 H.P. up to 0.95 for large ones. The mechanical efficiency of the screw = Useful work of axial thrust ÷ Shaft performance = 0.6 to 0.7 for best conditions. Diam. of Screw in ft., $D = K\sqrt{\text{I.H.P.} \div (0.01PN)^3}$; Total area of blades (developed) $= K_1\sqrt{\text{I.H.P.} \div N}$; P varies from $0.9D$ to $1.5D$. Speed V is measured in knots (1 knot = 6,080 ft. per hr.).

	V.	K.	K_1.
Cargo Boats,	8–13	17–19	19–15.5
Passenger and Mail Boats,	13–17	19.5–21.5	15–12.5
Do., very fine lines,	17–22	21–23	12.5– 9
Naval Vessels,	16–22	21–23.5	11.5– 7
Torpedo Boats,	20–26	25	7– 6

The Apparent Slip (in per cent) $S = (C - V) \div 100C$, where $C = P \times 60N \div 6{,}080$. $S = -2$ to $+8$ for slow freighters, $= 8$ to 15 for passenger and mail steamers, $= 13$ to 20 for naval vessels, $= 20$ to 27 for small, fine-lined boats.

Strength of Blades:—The indicated thrust T (divided by the number of blades Z) acts at a distance $0.35D$ from the center of shaft and causes a bending moment B_m. $B_m = \frac{T}{Z}(0.35D -$ distance from c. of shaft to root of blade). For a parabolic segmental cross-section (length l, thickness h) oblique to axis, the Moment of Resistance $= 0.076lh^2$, and consequently $f = B_m \div 0.076lh^2$. f (safe) in lbs. per sq. in. = 7,800 for cast steel, = 5,700 for bronze, = 2,800 for C. I.

Thickness of blades at tips = 0.25 to 0.8 in. for bronze, and 0.6 to 1.2 in. for C. I., according to size of the screw.

Indicated Horse-Power of Engines. $\text{I.H.P.} = p_m La(2N) \div 33{,}000$, where a = area of low pressure cyl. in sq. in. p_m, the mean effective pressure, depends on the absolute boiler pressure p, and also on the number of expansions:

$$p_m = kpc\left(1 + \log_e \frac{1}{c}\right). \quad \text{where} \quad c = \frac{\text{vol. of steam admitted into h. p. cyl.}}{\text{l. p. cyl. vol.} \div \text{h. p. cyl. vol.}}.$$

k has the following values at ordinary speeds:

Compound-Engines,	0.65 to 0.7	(at higher speeds, 0.6 to 0.65).
Triple-Expansion,	0.55 " 0.6	(" " " 0.52 " 0.58).
Quadruple-Expansion,	0.52 " 0.54	

Total Number of Expansions ($= 1 \div c$): Compound, small boats, 5 to 6; do., freighters, 7 to 8. Triple-Expansion, torpedo boats, 5 to 7; do , naval vessels, 6.5 to 8; do , express and freight steamers, 8 to 10. Quadruple-Expansion, express steamers, 10; do., freight steamers, 11 to 13. Cut-off in high-press. cyl. is at about 0.7 stroke (0.6 stroke for slow boats).

Piston Speed and Revs. per Min.

	Speed, ft. per min.	R.P M.
Torpedo Boats,	1,000–1,200	300–400
Armored Vessels,	800–1,000	100–150
Express Steamers,	800– 950	75– 95
Large Passenger Steamers,	700– 900	70– 90
" Freight "	700– 800	70– 85
Small " "	600– 750	95–130
" passenger "	400– 600	150–200

Steam Velocities (ft. per min.). Main steam-pipes, 6,000–8,000; steam passages: h. p. cyl., 5,000–6,000; intermediate cyl., 6,000–7,000; l. p. cyl., 7,200–8,500. For exhaust take 80% of above values. For small engines these velocities may be increased 20%.

Cylinders. Thickness of walls (cast iron) $t=\frac{dp}{5.120+10p}+0.4$ in., where p = gauge pressure in lbs. per sq. in at h. p. cyl., and d = diam. of h. p. cyl. in in. (This value of t is for h. p. cyl. with or without jacket and also for intermediate and l. p. cyl. linings. Cylinders without linings should be 0.2 in. thicker to allow for reboring.)

Thickness of cylinder head $t_1=t$ (for cast iron, head ribbed) $=0.6t$ to $0.65t$ for cast steel. Diam. of cyl.-head studs $=t$; pitch of studs $=3t$, $5.5t$ and $6.5t$ for high, intermediate and low-pressure cylinders, respectively.

Thickness of cyl.-head flange $=1.2t$, width $=2.6t$ to $3.3t$.

Relief valves (for both heads) should have a diam. $=(\frac{1}{12}, \frac{1}{17}, \frac{1}{20})\times$ diam. of (high, intermediate, low-pressure cyl.). Valves should open at about 8 lbs. above p.

Pistons. (Cast steel, coned, concave toward crank). Thickness near center, $t=0.0043d\sqrt{p}+c$; thickness near rim $=0.5t$ to $0.7t$.

$c=0.24$ in., 0.36 in., 0.48 in., respectively, for h., i. and l. pres. cyls.

p = boiler pressure in lbs per sq. in. for h. p. cyl., $=0.45\times$ boiler pressure for intermediate cyl., $=0.2\times$ boiler pressure for l. p. cyl. For forged steel take $\frac{7}{8}$ of above formula value for t.

Piston-Rods. (Medium hard steel, end tapered and fastened to head by nut.) Area at root of thread in sq. in. $\geq$ ($p\times$ area of h. p. cyl. in sq. in.) $\div$ 7,000. (For naval vessels and torpedo boats substitute 10,500 and 12,500 respectively for 7,000). Full section of rod beyond taper $=2\times$ area at root of thread.

Connecting-Rods. Length $=(2$ to $2.25)\times$ stroke. Diam. at piston end = diam. of piston-rod, approx.; diam. at crank-end $=(1.1$ to $1.4)\times$ diam. of piston-rod, according to length.

Bearings. The crank bearing is lined with white metal of a thickness $=(0.025\times$ diam. of bearing$)+0.2$ in. Thickness of cast-steel bearing cap at the middle $=(0.17$ to $0.24)\times$ diam. of bearing. Shaft bearing: thickness (cast iron) $=0.12d+0.2$ in.; for bronze, thickness $=0.09d+0.12$ in. Thickness of white-metal lining $=\left(0.2+\frac{d}{35}\right)$ in. d = shaft diam. in in.

Crank-Shafts (forged steel): $d^3=\frac{16}{\pi}\cdot\frac{T_m}{f}\cdot\frac{1}{\left(1-\frac{d_1^4}{d^4}\right)}$, where d = outer diam. of shaft in in. (d_1 = inner diam. in case of a hollow shaft), T_m = turning moment in inch-lbs. $=63,025\times$ I.H.P. $\div N$.

f_{safe} (average) in lbs. per sq. in. $=6,600$ for torpedo boats, $=5,700$ for naval vessels, $=4,500$ for mail steamers, $=4,000$ for freighters (max. and min. values are equal to average values $\pm 10\%$).

Crank-Throws. Outline described in part by circles (of diam. $=2d$) from centers of shaft and crank-pin, connected with filleting curves of radii $=d$. (d = diam. of shaft). Thickness of throws $=0.6d$ to $0.7d$. The shaft is enlarged $\frac{1}{40}$ of its diam. in the throw. Thickness of flanges on shaft $=0.25d$ to $0.28d$. Length of bearing $\div$ diam. of shaft $=1.4$ to 1.6 for torpedo boats, $=1.1$ to 1.4 for naval vessels, $=0.9$ to 1.2 for other vessels.

Surface Condensers. Cooling surface in sq. ft. required per I.H.P.: Compound, 5 to 6; triple-expansion, 3.5 to 5; quadruple-expansion, 3.5 to 4.6; torpedo boats, 26 to 32. (The lower values given are for naval vessels.) Condenser tubes are of brass, tinned inside and out, $\frac{5}{8}$ to $\frac{3}{4}$ in. outside diam. and about 0.04 in. thick.

Air-Pumps for Surface Condensers (Single-acting). Volume $=c\times$ vol. of l. p. cyl. $c=\frac{1}{14}$ to $\frac{1}{18}$ for compound; $=\frac{1}{20}$ to $\frac{1}{24}$ for triple-expansion; $=\frac{1}{24}$ to $\frac{1}{28}$ for quadruple-expansion. For injector condensers, Vol. $=(\frac{1}{6}$ to $\frac{1}{4})\times$ vol. of l. p. cyl.

Surface Condensers of High Efficiency. By passing the condensing water several times through the tubes (arranged in groups), and by providing for the thorough drainage of the water of condensation so that the tubes are not continually subjected to showers of water particles which

impair the surface contact, Prof. R. L. Weighton has designed condensers to be used in connection with dry air-pumps which condense 20 lbs. of steam per hour per sq. ft. of surface, condensing water required being 24 times the amount of feed-water used. He has effected a higher surface efficiency—36 lbs. per hour per sq. ft.,—but the condensing water required in this case is equal to 28 times the feed-water. Vacuum in both cases is 28.5 in. of mercury, feed-water temp. at inlet = 50° F. For a system with tight piping, capacity of air-pump = 0.7 cu. ft. per lb. of steam condensed per hour. The condenser tubes are provided with triangular wooden cores in order that the water may meet the tube surface in thin streams. Temp. of hot-well may be 3° to 5° higher than that corresponding to vacuum (up to 29 in.).

Circulating Pumps (Double-acting). Vol. = 0.025 × vol. of l. p. cyl. (approx.).

Boiler Accessory Apparatus.

Feed-Water Heaters. Let t and T = initial and final temperatures of water in degs. F. [average temp. = $(t+T)\div 2$]. B.T.U. transmitted per sq. ft. of surface per hour, per degree difference of temp. = c = 180 for water-tubes, 200 for coils, and 114.5 for steam-tubes (usually 2 in. diam.). Let T_s = temp. of exhaust (= 212° F. generally); then, B.T.U. per hour per sq. ft. $= c[T_s - 0.5(t+T)]$; lbs. steam condensed per sq. ft. per hour $= c[T_s - 0.5(t+T)] \div 966$. Velocity of water in tubes in ft. per min.: single-flow, 8.33; double-flow, 12.5; coils, 140. Sectional area within shell = $c \times$ total cross-section of tubes, where c = 6.3 to 9 for water-tubes, = 7.5 to 10 for steam-tubes,—the higher values for variable loads. For coil heaters, sectional area within shell = (11 to 8) × cross-section of exhaust pipe, inversely according to the capacity of heater. Open heaters with trays or pans: Volume of shell in cu. ft. = Capacity in H.P. $\div c$, where c = 2.2, 6, and 8 for very muddy, slightly muddy, and clear water respectively. Tray surface in sq. ft. = lbs. water heated per hr. $\div c$, where c = 118, 166, and 500 for very muddy, slightly muddy, and clear water respectively. These values for tray surface are for vertical heaters; for horizontal type of heater the values of c are about 8% lower.

Siphon or Barometric Condensers operating on the principle of injectors: Diam. of exhaust pipe in in., $d = \sqrt{c \times \text{lbs. steam to be condensed per min.}}$, where c = 0.81 when wt. of steam is less than 20,000 lbs. per hour (= 0.63 if greater than 20,000 lbs. per hr.). Diam. at throat in in. $= \sqrt{Ww \div 17{,}210}$; width of annular opening through which water is admitted $= Ww \div 39{,}550d$ (W = lbs. steam to be condensed per hr., w = lbs. water required to condense 1 lb. of steam).

Air-Pumps for Stationary Engines. Single-acting: vol. in cu. ft. $= 0.032S \div N$; double-acting: vol. in cu. ft. $= 0.016S \div N$. S = lbs. of steam condensed per hour, and N = r.p.m. (Ing. Taschenbuch.)

Locomotives.

Elevation of Outer Rail on Curves. E (in ft.) $= 0.06688GV^2 \div R$, where G = gauge of track in ft., V = velocity of fastest train in miles per hr., and R = radius of curve in ft. (*R. R. Gazette*, 3–16–'06.)

Combustion.

Natural-Gas Fuel for Steam-Boilers. The same economy is exhibited with a blue flame as with a white or straw-colored flame, but the latter affords greater capacity. One boiler H.P. may be expected from 43 to 45 cu. ft. of gas (at 60° F. under a pressure of 4 oz. *above* 29.92 in. of mercury). Fuel costs are the same with natural gas at 10 cents per 1,000 cu. ft. and semi-bituminous coal at $2.87 per ton of 2,240 lbs. (J. M. Whitham, A. S. M. E., Dec. '05.)

Efficiency of Combustion. The higher the percentage of CO_2 in the gases escaping into the chimney, the higher will be the efficiency of the furnace, and the production of CO_2 may be forced until the presence of CO indicates incomplete combustion. In good furnaces 10 to 15% of CO_2

may be realized. The approximate fuel loss (in per cent) due to incomplete combustion $=0.4(t_2-t_1)\div$(per cent by volume of CO_2), where $t_2=$ temp. of chimney gases and $t_1=$ temp. of air entering the furnace (both in degs. F.). An instrument called a CO_2 recorder indicates and records continuously the percentage of that gas present.

Mechanical stokers do not accomplish any marked saving of fuel over careful hand firing in plants where less than 200 tons of coal are used per month, but they yield much better results than average hand firing, are easily forced, maintain a uniform steam pressure, and assist greatly in the smokeless combustion of soft coals. They are adaptable to all kinds of solid fuels, and in this respect promote economy, for it often happens that a cheap, low-grade fuel may be employed, whereas with hand-firing a more expensive quality would have to be used.

Incrustation and Corrosion.

Boiler Purges. Caustic soda and lime-water combine with the carbonic acid contained in water (in combination as bicarbonates) and precipitate calcium and magnesium carbonates. Soda ash acts on the bicarbonates of lime and magnesia, forming bicarbonate of soda, which is decomposed by heat into CO_2 and sodium carbonate, the latter being precipitated.

Sodium aluminate and sodium fluoride are also used in waters containing bicarbonate of lime.

Trisodium phosphate is used where water contains sulphate of lime, precipitating sodium sulphate and calcium phosphate.

Internal-Combustion Engines.

Gas Producers are closed furnaces in which the fuel is burnt with a limited supply of air and steam, resulting in the production of gas. The air and steam are either forced (pressure producer) or drawn (suction producer) through a bed of incandescent coal or coke. The O of the air first combines with the C of coal to form CO_2. This passes up through the incandescent coal and changes to CO. When steam is mixed with the air and meets the burning fuel, H is liberated and the O of steam combines to form more CO. These, with the N of air and the volatile part of the fuel (CH_4) make up the resulting fuel-gas. Theoretically the best temperature is about 1,900° F. 1 lb. of coal with upwards of 0.7 lb. steam will yield from 65 to 75 cu. ft. of gas (135 to 140 B.T.U. per cu. ft.). Pressure producers are used for engines of over 200 H.P. In these the air and steam are furnished under a pressure of from 2 to 8 in. of water. The hot gas passes through an economizer where it preheats the air used and also gives up heat for the generation of the steam required. It then passes through the scrubber (vessel provided with trays of coke upon which water streams from above) and thence to the purifier (another vessel provided with trays of sawdust, and also with oxidized iron-filings when sulphur is to be removed from the gas). The best results are obtained from anthracite (pea size or larger) having less than 10 to 15% of ash and but little moisture. If the fuel contains more than 5 to 8% of volatile matter, it will cohere and prevent proper working of producer. Coal with an excessive amount of ash tends to choke up the air-passages.

Grate surface per H. P. = 6 to 8 sq. in. (the latter for producers of less than 25 H.P. capacity). The volume of producer per H.P. = 0.11 cu. ft., approx. (firing intervals of 3 to 4 hours), for anthracite, and 0.18 cu. ft. for coke. Vol. of scrubber = 0.9 to 1.1 cu. ft. per H.P. Vol. of purifier = 0.36 cu. ft. per H.P. In ordinary generators about 85% of the heat of the fuel leaves the producer, a loss of 15 to 20% being due to heating, radiation, and unburnt residue. Efficiency, 65 to 75%.

Suction-Producer Tests of a number of plants in London using Scotch anthracite (pea) showed a consumption of 0.85 to 1.1 lbs. per B.H.P. hr. for full load, and 0.9 to 1.25 lbs. at half load (larger values for 8 H.P., smaller for 20 H.P.). Volume of producers in cu. ft. per H.P. = 0.23 (for 20 H.P.) and 0.26 (8 H.P.). R.P.M., 200; mechanical efficiency, 81 to 84% at full load (69 to 71% at half load). M.e.p. about 79 lbs.

Blast-Furnace and Coke-Oven Gases. For each ton of iron smelted about 88,000 cu. ft. of blast-furnace gas are liberated. One ton of coal in coking yields about 8,800 cu. ft. of coke-oven gas.

Hot Tubes require from 5 to 7 cu. ft. of illuminating gas per hour (or 0.22 to 0.33 lb. gasoline per hour).

Denatured Alcohol is ethyl alcohol rendered unfit for consumption as a beverage by the addition of wood-alcohol, benzol, etc. It burns smokelessly with a hot, non-luminous flame, and the products of its combustion do not yield an unpleasant odor unless the percentage of benzol is excessive.

Ethyl alcohol (C_2H_6O) is made by the fermentation of sugars or starches contained in molasses, corn, potatoes, etc., with which malt and yeast are combined.

$$\begin{array}{lllll} \text{Starch} & +\text{Water} & =\text{Dextrose} & =\text{Alcohol} & +\text{Carb. dioxide.} \\ (C_6H_{10}O_5)_3 & +H_2O & =C_6H_{12}O_6 & =2(C_2H_6O) & +2(CO_2) \end{array}$$

The alcohol is distilled off by the proper application of heat, absolute alcohol (100%) being that in which no water is present.

Specific Gravity of Ethyl Alcohol at 59° F. Approx. Lower Heating Value.

% by vol.	% by wt.	Sp. g.	
100	100	.7946	11,700 B.T.U. per lb.
95	93.8	.805	10,900 " " "
90	87.7	.815	10,100 " " "
One gal. abs. alcohol = 6.625 lbs.			77,500 " " gal.

Composition of absolute alcohol $= 0.522C + 0.13H + 0.348O$. Air required for combustion of 1 lb. alcohol = 9 lbs., or 111.5 cu. ft. at 62° F.

Boiling-point = 173.1° F. freezing-point = −200° F.

Specific heat of liquid at 32° F. = 0.5475. k_p of vapor = 0.4534; $kv = 0.4$; $k_p \div kv = n = 1.14$. Law of compression: $PV^{1.14} = P_1V_1^{1.14}$.

Alcohol motors are started up with gasoline, and, when warmed up sufficiently, alcohol vapor is used. Cooling water required is about 20 lbs. per H.P. hour, and efficiency is promoted by having its temperature as high as possible.

Denaturants.

		Sp. g. (59° F.)	Boiling-point.	Lower Heating Value (Approx.).
Methyl (or Wood-alcohol)	CH_4O	0.800	151° F.	8,300 B.T.U. per lb.
Benzol	C_6H_6	0.866	176°	17,200 " " "
Acetone	C_3H_6O	0.800	133°	12,600 " " "
Pyridine	C_5H_5N		117°	17,000 " " "
Gasoline		0.700	180–210°	19,000 " " "

Denatured Alcohol Mixtures [parts by volume added to 100 vols. of 90% (vol.) alcohol].

	Sp. g.	CH_4O	C_5H_5N	C_3H_6O	C_6H_6	Gasoline
French	0.832	7.5		2.5		
German	0.819	1.5	0.5	0.5		0.5
Do., "Motor Spirit"	0.825	0.75	0.25	0.25	2	

No more heat should be used than is necessary to vaporize the mixture, high temperatures limiting the allowable compression and decreasing the economy. For a 90% (vol.) alcohol 7.9 lbs. of air are theoretically required for the combustion of 1 lb. Assuming 11.8 lb. (an excess of 50%) in practice, 1 cu. ft. dry air (at 60° F) is supplied for 0.0065 lb. alcohol, or as 90% (vol.) = 87.7% (wt.), 1 lb. air will carry 0.877 (1 ÷ 11.8) = 0.0075 lb. of abs. alcohol, and (1 − 0.877)(1 ÷ 11.8) = 0.01 lb. of water If the air be considered as saturated with moisture when entering the vaporizer at 60° (26 in. mercury), it will contain 0.013 lb. water in addition to the 0.01 lb. in the alcohol, or 0.023 lb. in all. A temp. of 77° F.

will vaporize this amount of water and also 0.162 lb. of alcohol, consequently the smaller amount of alcohol actually used will be superheated.

Under these conditions (total heat of vaporization at 77° F. being 458 B.T.U. per lb.) the heat required for vaporizing is about 6% of the heating value of the alcohol and may be obtained from the exhaust, or by preheating the air used to about 270° F.

The best results are obtained by compressing the mixture to 180–200 lbs. per sq. in., the corresponding max explosion pressure being about 450 lbs. per sq. in.

90% (vol.) alcohol costs about 15 cents per gal. (2.21 cents per lb.) when made from good corn at 42.4 cents per bushel. To compete with gasoline at 15 cents per gal. its cost must be reduced to 12 cents per gal., which is possible through the use of low-grade grain, cheap vegetable matter, and refuse containing sugar or starch.

	Gasoline.	Kerosene.	Alcohol (90% vol.).
B.T.U. per lb.	19,000	18,500	10,100
Cost per gal. in cents	15	13	15
" " lb. " "	2.57	1.88	2.21
Specific gravity	0.710	0.8	0.815
* Lbs. per B.H.P. hour	0.58	0.725	0.803
* Thermal brake efficiency in per cent.	23.	18.	31.7
B.T.U. per B.H.P. hour	11,000	14,140	8,030
Fuel cost per B.H.P. hour (cents)	1.485	1.446	1.758

Gas-Engine Design.

Pistons. Max. pressure on piston, $P=0.7854pd^2$. Permissible surface pressure, $k=18$ to 22 lbs. per sq. in. (frequently as low as 8 lbs. where length of piston is unimportant). Length of piston $l \geq 0.11P \div dk$. Generally, $l=2.25d$ to $2.5d$ for small engines ($=1.25d$ to $1.5d$ for large engines). Wrist-pin diam. $d_1=\sqrt[3]{pd^2l_1 \div 5{,}680}$, where l_1=total length of pin$=0.75d$; bearing length of pin is about $0.5d$. Thickness of piston wall$=0.02d$+depth of packing-ring groove+0.2 in. To provide for expansion the piston is tapered from d at the crank-end to from $0.995d$ to $0.998d$ at head end. Pistons over 8 in. in diam. have from 4 to 6 radial stiffening ribs.

Piston-Rings. Radial depth, $s=0.022d$; width, $b=0.028d$ to $0.044d$. No. of rings$=d \div 5b$. Space between grooves$=b$; depth of groove$=s+(0.02$ to 0.08 in.).

Cylinders. Thickness of walls for strength, $t=[0.42pd \div (f-p)]$, where f for C.I. may be as high as 3,500 lbs. per sq. in. If $d>24$ in., the wall may be gradually tapered from t at compression end to $0.5t$. To allow for reboring, etc., 0.16 in. to 0.4 in. should be added to t throughout the length. Jacket: where axial forces do not enter into consideration, t_1 of jacket≥ 0.4 in. If the jacket is cast in one piece with the cylinder, $t_1=0.022(\bar{d}+2t)$ for a test pressure of 420 lbs. per sq. in. (corresponding to $f=8{,}500$ lbs. per sq. in. in a cold test).

Valves. h_1=lift in in.; d_1=diam. in in.; $a_1=\pi d_1h_1$=area of valve opening in sq. in.; a=piston area in sq. in.; S=stroke of piston in ft.; c=mean velocity of piston in ft. per sec.; v=mean velocity of gas through valve in ft. per sec.; d=diam. of cyl. in in. Then, $a_1=ac \div v$, and, if $h_1 \leq 0.25d_1$, $\pi d_1h_1=\pi d^2NS$, or $d_1h_1=d^2NS \div 1{,}200$. v (mean)$=82$ ft. per sec. If l of connecting-rod$=2.5S$, $v(\text{max.})=1.6v=131$ ft. per sec. In order not to exceed this velocity each position of the piston requires a corresponding lift of the valve, $h_1 \geq d^2NS\psi \div 9{,}840d_1$, where $\psi=\sin\alpha(1\pm\lambda\cos\alpha)$, α being the angle made by the crank and the direction of center-line of piston-rod.

* Best results obtained.

If $\lambda = 0.5S \div l = 0.2$,

% stroke, outward,	2	5	10	20	30	40	45	5
% " return,	98	95	90	80	70	60	55	5
$\psi =$	.304	.472	.648	.853	.962	1.011	1.018	1.01
% stroke, outward,	55	60	70	80	90	95	98	100
% " return,	45	40	30	20	10	5	2	0
$\psi =$	1 00	.976	.892	.759	.554	394	.251	.0

Thickness of valve in in. $= \sqrt{pd^2 \div 25,600}$, where d = outside diam of valve. Diam. of valve-seat $= 0.98d - 0.32$ in. Diam of valve-stem $= 0.125d + (0.2$ in. to 0.32 in.). Spring tension on valves: for throttling regulation, not less than 7 lbs. per sq. in. of cone surface; for automatic valves, from 0.7 to 1.00 lb. per sq. in. of cone surface, according to speed.

Fly-Wheels. Weight of rim in lbs. $= 2,165,320kK(0.75+\rho) \cdot$ I.H.P. $\div v^2N$, where ρ = m.e.p. on compression stroke ÷ m.e.p. on power stroke $= 0.3$ usually; k has the values given on page 74; v = mean vel. of rim in ft. per sec., N = r.p.m. and K has the following values:

	4-cycle.	2-cycle.
One cylinder, single-acting,	1.000	0.400
" " double-acting,	.615	.110
Two cylinders, twin, single-acting,	.400	.400
" " single-acting, cranks 180° apart,	.645	.085
" " double-acting, tandem or 4 twin opposing cyls.,	.085	—

Total weight of wheel is about 1.4 times wt. of rim. (The foregoing matter has been derived chiefly from Güldner's "Verbrennungsmotoren.")

Proportion of Parts. It is now customary to assume an explosion pressure of 450 lbs. per sq. in. (m.e.p. = 70 lbs.) and a mean piston speed of 800–850 ft. per min. For this pressure the values given on pages 99 and 100 should be altered to the following:

t of cyl. wall $= 0.092d + 0.25$ in.; outside diam of cyl-head studs $= 0.29d\sqrt{1 \div \text{No. of studs}}$; l of piston $= 2.25d$; t of rear piston wall $= 0.12d$; wrist-pin: length $= 0.47d$; diam. $= 0.27d$; connecting-rod diam. at mid-length $= 0.29d$; crank-pin: diam. $= 0.47d$, length $= 0.52d$; crank-throws: thickness $= 0.3d$, width $= 0.63d$; crank-shaft (at main bearings): diam. $= 0.43d$, length $= 1.12d$.

Expansion must be allowed for between the jacket and cylinder walls. (For 144° F. increase in temp., a cyl. 60 in. long will expand 0.053 in. in length.)

Large Gas-Engines (over 200 to 300 H.P.) should be double-acting, tandem, in order to obtain maximum power with minimum weight. (Junge, Power, Dec. '05.)

Marine Gas-Engine (Otto-Deutz). 4-cyl. horizontal (20–25 H.P per cyl.); 275–325 r.p.m.; cylinder: diam. = 10.8 in., length = 33.72 in.; stroke = 15.6 in.; crank-pin: length = diam. = 5.4 in.; length of connecting-rod = 2.25 × stroke; crank-throws: 6 in. wide × 3.7 in. thick; diam. of wrist-pin = 2.8 in.

Gas Turbines. The best results are obtained with high compression, rapid introduction of heat (around 900 B.T.U. per lb.), and by an exhaust temp. of about 1,300° F. absolute. The charge should be compressed to about 570 lbs., maintained at about 140 lbs. in combustion-chamber, and exhausted at or below atmospheric pressure. Velocity at nozzle varies from 1,600 to 2,600 ft. per sec. according as the temp. of combustion ranges from 1,800° to 4,500° F., absolute. For a temp. of 3,600° F. abs. (compression 350 lbs.), the sectional area of combustion-chamber = 100 × sectional area of nozzle, and vol. = sectional area × 5 to 10 times the diam Nozzles to resist heat are made of corundum, metal-tipped. Peripheral speed of wheels should not exceed 650 ft. per sec. Wheels and vanes should be made of nickel steel, which is not weakened or unduly oxidized by the temperatures employed. (L. Sekutowicz, Mem. Soc. des Ingenieurs Civils, France.)

Compressed Air.

Blowers and Compressors.

	Pressures employed lbs. per sq. in. (gauge).	Capacities (cu. ft. per min.)
For blast-furnaces,	4 to 10 lbs.	Up to 65,000
" Bessemer converters,	15 " 45 "	" " 30,000
" compressed-air transmission,	70 " 115 "	" " 10,000
" " reservoir storage	1,000 and upwards	
" " " "	2,000 (for torpedo boats)	

For pressures above 75 lbs., two- or three-stage compression should be employed, the air passing from compression cylinders into intercoolers, where it is split up into thin streams and flows over the surfaces of tubes chilled by water circulating through them. For two-stage compression, pressure in intercooler $=\sqrt{\text{final pressure to be obtained}}$. For three-stage compression (high pressures, 1,000 lbs. and over), pressure in first intercooler $=\sqrt[3]{\text{final pres.}}$; pressure in second intercooler $=\sqrt[3]{\text{square of final pres.}}$

The mean piston velocities employed range from 400 to 600 ft. per min. Blowers for blast-furnaces have strokes of from 3 to 6 ft., and r.p.m. up to 50. Air and steam cylinders are generally of equal dimensions and have the same length of stroke. $pm(\text{air})=\eta pm(\text{steam})$. For large, horizontal blast-furnace blowers $\eta=0.85$, for blowers for converters and compressors $\eta=0.75$ to 0.85 (η = mechanical efficiency).

The volumetric efficiency ranges from 90 to 95%. It may be determined from the low-pressure cyl. diagram: Volumetric efficiency = length of card on atmospheric line ÷ total length between the extreme end ordinates of card. Velocity of flow through valves = 3,000 to 5,000 ft. per min. (suction), = 5,000 to 7,000 ft per min. (compression).

I.H.P. $=144cxQ(p-14.7)\div(0.9\times33{,}000)$, where $c=1.3$ to 1.4 for blast-furnace blowers, $=1.35$ to 1.5 for compressors and blowers for converters; Q = cu. ft. of air per min.; p = absolute pressure of air in lbs. per sq. in.; 0.9 = specific weight of air at 29.52 in. of mercury and at 77° F. compared with air at 29.92 in. of mercury and at 32° F.

Values of x:

For $p=$	25	50	75	100	125
x (poor cooling) $=$	.81	.61	.50	.44	.40
x (efficient cooling, compression according to $pv^{1\cdot25}$) $=$	.77	.57	.46	.40	.35

Ft.-lbs. of work theoretically required to compress 1 cu. ft. of free air from p to $p_1=(3.44\times144p)\left[\left(\frac{p_1}{p}\right)^{0\cdot29}-1\right]$ (see page 102).

Rotary Blowers consist of two impeller wheels revolving in a close-fitting casing with equal velocities and in opposite directions, the air being drawn in at right angles to the axes of impellers and delivered compressed at the opposite opening. The profiles of the impellers are developed in the same manner as are the teeth of gear-wheels.

Capacity in cu. ft. per sec., $q=\lambda N\pi B(D^2-A)\div(4\times30)$, where N = r.p.m.; B = axial length, and D = diam. of impellers, both in feet; A = area of cross-section of impeller in sq. ft.; λ = volumetric efficiency = 0.6 to 0.95. Mechanical efficiency ranges from 0.45 to 0.85. Pressures from 12 to 80 in. of water (0.43 to 2.9 lbs. per sq. in.).

Mechanical Refrigeration.

Plate Ice vs. Can Ice. Plate ice does not require the use of distilled water in its production. 1 lb. of coal will make about 10 lbs. of plate ice, some 275 sq. ft. of freezing surface being required per ton capacity.

In the manufacture of can ice filtered or distilled water must be used, otherwise the impurities contained in ordinary water will be retained in the core of the block. Can ice does not keep well when stored. 1 lb. coal will make from 6 to 7½ lbs. of can ice. Plate systems cost from 40 to

75% more than can systems. (For 50-ton plant, a can system costs about \$550 per ton capacity).

Heating and Ventilation.

Heat Losses due to conduction and radiation, H (in B.T.U.) = Equivalent glass surface, $E \times (t+15°)$, where t = difference between temp. of room and outside temp. = 70° F., generally.

$$E = \frac{\text{Exposed wall surface}}{4} + \text{Glass surface} + \frac{\text{Exposed ceiling or floor surface}}{20}$$

(Surfaces in sq. ft.) Exposed surfaces are those one side of which is subjected to temp. of outside air.

To H must be added, $V = \frac{nct}{55}$ to provide for ventilation losses, where n = No. of changes of air per hour, c = contents of room in cubic ft. The total loss $(H+V)$ must be increased 15% for E. exposures and 25% for N. and W. exposures.

Hot-Air Heating. Air should be heated to about 140° F. No. of cu. ft. of air heated from 0° to 140° = Q = total heat loss in B.T.U. ÷ 2.87. Assuming that 5 lbs. of coal are burnt per sq. ft. of grate-area per hr., and that each lb. supplies 8,000 B.T.U., area of grate in sq. ft. = $Q \div 14{,}000$. The heating surface of furnace should be from 12 to 20 times the grate area, 1 sq. ft. of heating surface giving off about 2,500 B.T.U. per hr. The fire-pot should not be less than 12 in. deep, and the cold-air box should have an area of about 75% of the combined cross-section of all the pipes. For an average outside temp. of 25° F., from 1.75 to 2 lbs. of coal are burnt per hr. per sq. ft. of grate area For temp. of −5° F., from 4 to 4.5 lbs.

Area of Pipes for Hot-Air Heating. Volume of air in cu. ft. per min. $V = E(t+15) \div (60 \times 1.1)$. Velocities of air, v = 280, 400, and 500 ft. per min. for 1st, 2d, and 3d floors respectively. Area of pipes in sq. ft. = $V \div v$, or, diam. of pipe in in. = $\sqrt{184 V \div v}$. Area of air outlets should exceed 1.1 × grate area. Area of registers = 1.25 × area of pipe supplying same. (Condensed from Proceedings Am. Soc. Htg. and Vent. Engrs., W. G. Snow and I. P. Bird.)

Blower System of Heating and Ventilating. In this system the air is blown by means of a fan over coils of pipe through which steam circulates. Cu. ft. of air required = Total B.T.U. required ÷ 55(140 − 70), where 140 = degs. F. air is to be heated, and 70 = degs. F. temp. to which rooms are to be heated. The coils are generally of 1-in. pipe, from 200 to 250 linear ft. of pipe being used per 1,000 cu. ft. of air to be heated per min. Air velocities (ft. per min.): Mains, 1,500–2,000; branches to register flues, 1,000–1,200; flues to registers, 500–700; from registers, 300–500.

Steam Heating, Sizes of Mains for. (Indirect Radiation.)
Sq. ft. of radiating surface supplied by pipe 100 ft. long = A.

$A = (82 + 2.3\,p)d^{2.61}$, where $p < 16$ lbs. (¼ lb. allowed for drop).
$A = (138 + 2.15p)d^{2.61}$, " $p > 16$ " (½ " " " ")

For other lengths, multiply by factor c:

L in ft.	=	50	200	400	600	800	1,000
c	=	1.4	.7	.51	.41	.35	.31

(p = abs. pressure in lbs. per sq. in.; d = diam. of pipe in in.)

Diam. of returns, $d_1 = 0.5d$ when $d > 7$ in. If $d < 4$ in., d_1 is one size smaller; if d = 4 to 7 in., $d_1 = 3\frac{1}{2}$ in.

Direct Radiation: For W.I.-pipe radiators, A will be 20% greater than above for a given diam. d, and for C.I. radiators 30% greater.

[The foregoing has been digested from matter contained in *The Engineer* (Chicago) for Jan. '06.]

Compare with: Square feet of radiating surface = lbs. steam per min. × 145 (= lbs. steam per min. × 60 min. × 966 B.T.U. per lb. ÷ 400 B.T.U. radiated per sq. ft. per hour). See also formulas on page 70 for Flow of Steam in Pipes.

Cooling of Hot-Water Pipes. Ordinary 2-in. pipes (0.154 in. thick) with water at 140° F. cooling to 32° F. (air about 7° F.) lose approximately as follows:

0.55 B.T.U. per sq. ft per hr. per degree drop in temp (still air).
1.05 B.T.U. per sq. ft per hr. per degree drop in temp. (air moving 1 ft. per sec.).
1.5 B.T.U. per sq. ft. per hr per degree drop in temp. (in still water at 32° F.).
4.5 B.T U. per sq. ft. per hr. per degree drop in temp. (in water moving 1½ in. per sec.).
(*Power*, Feb. '06.)

HYDRAULICS AND HYDRAULIC MACHINERY.

Plunger-Pumps. Strainer area = (2 to 3) × cross-section of suction-tube. Area of valve-passages = (1.5 to 2) × cross-section of suction-tube. Valves should be of pure rubber.

Suction air-chamber vol. = (5 to 10) × vol. of pump cyl. Suction velocity = 150 to 200 ft. per min. Vol of pressure air-chamber = (6 to 8) × vol. of pump cyl.

Pressure velocity = 200 ft. per min. for large pumps and long pipes, = 300 to 400 ft. per min. for small pumps and short pipes.

Thickness of cyl. wall = $0.02d + 0.4$ in. for vertical pumps (for horizontal pumps make thickness 25% greater).

Thickness of air-chamber walls, $t = 0.42pd \div (f_t - p)$, where p = lbs. per sq in., gauge, f_t (safe) = 2,100 for C. I. = 8,500 to 10,000 for W. I.

Efficiencies up to 93%, usually 80 to 85%.

Centrifugal Pumps. Outer rim velocity in ft. per sec., $v_1 = 2\pi r_1 N \div 60$; relative discharge velocity, do., $= v_d = \phi v_3$ (v_3 = entering velocity of water). $\phi = r_2 b_2 \div r_1 b_1 \sin \alpha$. ($r_1$, b_1, and r_2, b_2 = outer and inner radii of wheels and vane widths, respectively; α = angle included between tangent to wheel (in direction of motion) and direction of end element of a vane, produced).

Theoretical pressure height, $H_1 = (v_1^2 + v_d v_1 \cos \alpha) \div g (= 1.3H$ for short conductors and $1.5H$ for average lengths). H = total height of delivery = suction head + pressure head. Head against which pump can lift = $(v_1^2 - v_2^2) \div 2g$. $r_1 = 2r_2$ (diam. of suction-tube is made equal to r_1); $v_3 = 3$ to 10 ft. per sec. No of vanes = Z = 6 to 12. Efficiency of best pumps is around 80%.

Cu. ft. of water pumped per sec. $= \left(2\pi r_2 - \dfrac{Zt}{\sin \alpha_1}\right) b_2 v_3$, where α_1 = angle between tangent to vane at inner end, and tangent to inner circle of radius r_2; t = thickness of vane in ft.

Pumping-Engines. Area of valve-seat openings = area of plunger × plunger speed in ft. per min. ÷ 200. (Chas. A. Hague.)

SHOP DATA.

High-Speed Steel Practice (Speeds in ft. per min., cuts in in.).

	Light		Heavy	
	Speed.	Cut.	Speed.	Cut.
C. I., medium,	75	$\frac{1}{32} \times \frac{1}{32}$	47	$\frac{1}{2} \times \frac{1}{8}$
C. I. (hard), tool-steel	35	"	20	$\frac{1}{4} \times \frac{1}{8}$
Steel, soft,	150	"	67	$\frac{1}{2} \times \frac{1}{8}$
" hard,	92	"	50	$\frac{3}{8} \times \frac{1}{8}$
Mall. iron,	100	"	80	$\frac{3}{16} \times \frac{1}{8}$
Brass,	120	"	90	$\frac{1}{2} \times \frac{1}{8}$
Chilled iron	3 to 12 ft. per min., all cuts.			

The above values for turning are for diameters of work ≥ 6 in.; for smaller diams. use speeds 10 to 15% lower. For milling, multiply above speeds by 1.5,—for boring, multiply by 0.6 to 0.8.

Drilling: Average peripheral speeds (feeds 0.008 to 0.02 in. per rev. for drills > $\frac{1}{2}$ in.):

Material,	C. I.	Steel.	Mall. Iron.	Tool Steel.	Brass.
Speeds,	80	67	78	33	127

Reaming: Periph. speed = Periph. speed of drill of same size $\times 2 \div$ No. of lips on reamer. Feed for reamer $= \frac{1}{2}$(drill feed $\times$ No. of reamer lips).

Milling: Periph. speed of cutter for a cut $\frac{1}{8}$ in. deep, and a feed of 0.01 in. per tooth of cutter per rev.: C. I., 90; mall. iron, 86; soft steel, 75; tool steel, 37; brass, 140.

Planing: 50 ft. per min. for steel. (O. M. Becker, *Eng. Mag.*, Aug. '06.)

Turning:	Steel Shafting.	C. I.		Forged Steel.	
Ft per min.,	61	150	102	160	32–100
Lbs. per min.,	3.64	2.75	5.6	10	35
Milling:	Steel.			C. I.	
Cut,	$7\frac{1}{2} \times \frac{1}{2}$ in., 6 ft. per min.			$6 \times \frac{3}{4}$ in., 4 ft. per min.	
Lbs. per min.,	6.4			5	

Drilling: 50 to 100% higher speeds than given above by Becker. (Results with "A. W." steel; *Engineering*, London, 12–15–'05.)

Tool.	Material.	Ft. per min.	Lbs. per min.
Lathe,	C. I.,	106	2.63
"	Steel,	44	2.3 to 3.43
"	"	170	1.69
"	W. I.,	54	4.2
Wheel-lathe,	Steel,	14	6.
Planer,	Cast steel,	30	3.2
"	C. I.,	29	18.3
Shaper,	Brass,	120	2.03
Drill ($1\frac{1}{4}$ in.),	W. I.,	54	.88
Boring-mill,	Steel,	60	1.1

(G. M. Campbell, *Am. Mach.*, 1–25–'06.)

The average cutting force varies from 100,000 lbs. per sq. in. for soft C. I. to 170,000 lbs. for hard C. I. Very hard C. I. may be cut at 25 ft. per min.; above 125 ft. per min. for C. I., tools begin to wear rapidly. (Univ. of Ill. tests.)

H.P. Required by Machine Tools $= C \times$ lbs. removed per min. $C =$ 2.5 for hard steel, 2 for W. I., 1.8 for soft steel, and 1.4 for C. I. (G. M. Campbell, W. Soc. Eng'rs, Feb. '06.)

Standards for Machine Screws. (Threads U. S. form; adopted by the A. S. M. E.) $p'' =$ pitch $= 1 \div$ No. of threads per in.; $d =$ depth $= 0.612\,p''$; flat at top and bottom $= p'' \div 8$; $D =$ diam. of body of screw.

	Diam. of Head.	Thickness of Head, t.	Slot Width.	Slot Depth.
Round head	$1.85D - 0.005$	$0.7D$	$0.173D + 0.015$	$0.35D + 0.01$
Flat (counter-sunk)	$2D - 0.008$	$\dfrac{D - 0.008}{1.739}$	" "	$\dfrac{D - 0.008}{5.217}$
Flat fillister head	$1.64D - 0.009$	$0.66D - 0.002$	" "	$0.33D - 0.001$
Oval fillister head	$1.64D - 0.009$	$0.66D - 0.002$	" "	$0.44D - 0.001$

Height of oval fillister head $= 0.88D - 0.003$; radius of oval head = diam. of head. Included angle of flat head $= 82°$.

Diam. in in.	0.060	0.073	0.086	0.099	0.112	0.125	0.138
Threads per in	80	72	64	56	48	44	40
Diam. in in.	0.151	0.164	0.177	0.190	0.216	0.242	0.268
Threads per in. . . .	36	36	32	30	28	24	22
Diam. in in.	0.294	0.320	0.346	0.372	0.398	0.424	0.450
Threads per in. . . .	20	20	18	16	16	14	14

Force Fits. Pressure required in tons $= 786\,dl\delta \div d^{1.06}$, where $d =$ diam. of piece, $l =$ length, $\delta =$ allowance for fit, all in inches. (S. H. Moore.)

International Metric Threads. Angle of thread $=60°$. The top of thread is flatted off ($\frac{1}{8}$ of its height) and the bottom is rounded to $\frac{1}{16}$ its height, making total depth of thread $=\frac{13}{16}\times$ the depth of a sharp V thread of same pitch.

Cost of Electric Power.—In large street-railway power-houses (2,000 to 10,000 kw. capacity) with coal costing $3.50 per ton, the cost of one kilowatt hour at the switchboard is about $0.0078. (C. H. Hile, *Power*, Nov. '05.)

Miscellaneous Machine Design.

Power-Hammers. Lifting force $P=$ weight of hammer $W\times\alpha$, where $\alpha=1.2$ to 2. Lift $L=3$ to 6 ft., $W=100$ to 2,000 lbs. Velocity $=150$ to 250 ft. per min.; strokes per min. $=20$ to 30.

Steam-Hammers. $W=50{,}000$ to 250,000 lbs., $\alpha=1.5$ to 2. No. of strokes per min. $=72\div\sqrt{L}$. Greatest lift L, in ft., $=0.25\sqrt[3]{W}$. Diam. of piston-rod in in. $=0.055\sqrt{W}$. For small hammers ($W=150$ to 2,000 lbs.), $\alpha=2$ to 3.5.

Piston-rod diam. in in. $=(0.5$ to $0.65)\times$ piston diam.

Weight of Anvil and Base $W_1=cLW$; ($c=1.8$ for iron forging, $=3$ for steel-work)

Pressure exerted on anvil $=xLW+W_1$, where $x=18$ to 25 for iron-work, and 25 to 35 for steel.

Riveters are designed to furnish 100,000 to 200,000 lbs. pressure per sq. in. of rivet section (according to the hardness of rivets), and about one-third of this pressure for holding plates together while being riveted.

Bending Rolls. Diam. of roll $d=2\sqrt{bt}$, where $b=$ width of plate, and $t=$ thickness (d, b, and t in in.).

Punches. Diam. of punch $d_1=d$, or $d-\frac{1}{8}t$; diam. of hole in die $=d_1+\frac{1}{4}t$; ($d=$ diam. of hole in plate, $t=$ thickness of plate, both in in.).

Greatest force required $=\sigma\pi dt$. σ (or shearing strength of material in lbs. per sq. in.) $=84{,}000$ to 100,000 for steel plates, $=55{,}000$ to 85,000 for W. I. ($=17{,}000$ to 28,000 when heated to a dark red), $=35{,}000$ to 55,000 for copper, $=13{,}000$ to 20,000 for zinc. Velocity of stroke $=3$ to 4 ft. per min.

Shears. Vertical clearance of blades $=2°$; angle of cutting edge of blades $=75°$ (approx.). Angle included between cutting edges of both blades $=\alpha=8°$ to $10°$. Greatest pressure required (when $\alpha=0°$) $=\sigma bt$, where $b=$ width of blade and $t=$ thickness of plate to be sheared. Pressure required when $\alpha>0°=\dfrac{0.225\sigma t^2}{\tan\alpha}$. Cutting speed $=3$ to 6 ft. per min.

Circular Shears are used for cutting sheets up to 0.2 in. in thickness. Diam. of blades $=70\times$ thickness of sheets to be cut, circumferential speed $=100$ to 200 ft. per min.

Rolls for W. I. Diam. of roll in in. $d=(t_1-t_2)\div(1-\cos\theta)$, where θ is obtained from the relation. $\tan\theta=\mu$. μ for W.I. at rolling heat is approx. equal to 0 1, whence $d=(t_1-t_2)\times 200$. ($t_1=$ thickness of metal before rolling, $t_2=$ thickness after).

Planers. Speed for tables over 6 ft. wide $=12$ to 20 ft. per min.; for tables less than 6 ft. wide, from 20 to 28 ft. Return speed $=4\times$ cutting speed.

Shapers. Cutting speeds up to 48 ft. per min.; return speeds $=4\times$ cutting speed.

Belt-Conveyors. Rubber-covered belts from 8 to 60 in. wide running on rollers (3 to 5 in. in diam.) are used for conveying grain, coal, ashes, etc., where the angle of elevation is not over 23°.

Spacing of Rollers.

	Driving side.	Return side.
Grain	6 to 12 ft.	12 to 18 ft
Coal	4 to 6 "	8 to 12 "

For changing direction guide rollers 6 to 8 in. diam. are used; if the deviation is abrupt, rollers from 12 to 20 in. diam. are employed. The tension of belt is maintained by weights or a screw.

Belt Velocities V, in ft. per min.:

Bran, light grains, etc., 400; heavy grain, 500 to 600.
Coal (horizontal belt), 460; elevating, 660 to 900.
Sorting or gathering belts, up to 60.

Cubic feet moved per hour $=0.0224V(0.9b-2)^2$, where b = width of belt in in.

Screw-Conveyors consist of sheet-metal helicoids mounted on hollow shafts, with bearings 8 ft. apart for a 4-in. screw (up to 12 ft. apart for an 18-in. screw). Used where elevation angle is less than 30°.

Troughs of sheet metal 0.08 to 0.16 in. thick; clearance between screw and trough = 0.12 to 0.25 in. Spirals of rectangular-section steel bars wound edgewise and connected to shaft at about every 20 in. perform about 20% less work than screw conveyors.

Sections of spirals.	0.8×0.2 in.	1.5×0.28	2.5×0.28	3×0.28
Diam. of trough.	4 in.	8 in.	12 in.	20 in.

Diam. of screw $d \leqq 17$ in., generally. Pitch of spirals $=0.7d$. R.p.m. $=282 \div \sqrt{d}$.

If 42% of the cross-section of trough is assumed to be filled with the material to be moved, then, Cu. ft. moved per hr. $=2.265\sqrt{d^5}$.

H.P. required $=(0.061$ to $0.091)\times Lq\gamma$, where L = length of screw in ft., q = cu. ft. delivered per sec., and γ = lbs. per cu. ft. of the material moved.

ELECTROTECHNICS.

Storage Batteries consist of lead plates immersed in dilute sulphuric acid. These plates are either coated with a paste made of red lead (or red lead and litharge), or they are cast in the form of grids, the paste being forced into the holes of the grids under pressure. The number of negative plates is always one more than the number of positive plates. The H_2SO_4 must be pure (free from HNO_3, HCl, and Sb) and diluted only with distilled water, the acid being always poured into the water,—never *vice versa*. The dilute acid or electrolyte should have a sp. g. of about 1.14 (=19° Baumé) at the beginning of a charge, which rises to 1.18 to 1 2 (23° to 25° Baumé) at the completion of charge. The density becomes altered in use through evaporation of water, loss through ebullition, etc., and water or acid should be added from time to time to keep the plates covered with ½ to ¾ in. of the electrolyte. The sp. g. is the best guide to the condition of the cell. Voltage of cell = 2 volts, approx., at beginning of charge, rising slowly to 2.2 volts, thence more rapidly to 2.7 volts. Discharge begins at about 2 volts, quickly dropping to 1.97 volts, then slowly to 1.9 volts and then rapidly to 1.83 volts. If no current is taken from cell, its voltage is about 2 volts, regardless of the degree to which it is charged. Current strength varies (according to size and construction of cell) from 5.5 to 8.4 amperes per sq. ft. of plate area (charging), to 8.4 to 11 amp. per sq. ft. (discharging), or 1.1 to 1.3 amp. per lb. of plates.

Capacity is measured by the number of ampere-hours which a cell will yield up to a certain defined drop in voltage (7 to 20%) down to 1.83 volts. The capacity is greater the slower the discharge and varies from 1.8 to 3.6 amp.-hr. per lb. of plates (rapid discharge) to 5.5 to 7 amp.-hr. per lb. (slow discharge).

Efficiency:—Good cells yield from 90 to 95% of the amperage with which they are charged, and (the voltage of discharge being lower than that of charge) from 75 to 85% of charging energy in watts.

The first charge must be undertaken as soon as the electrolyte is poured into the cells and it should continue until the positive plates have a dark-brown color and the sp. g. of electrolyte has risen from 1.14 to at least 1.18. Time required: from 16 to 50 hours. Charging is generally accomplished with voltages up to 2.4 volts for a steady current, and is interrupted when gas bubbles slowly begin to form at about 2.25 volts (i.e , when violent ebullition occurs at about 2.5 volts) Cells should be fully charged when lying unused, and should be recharged every 10 days or so, if possible

The Dielectric Strength of Insulating Materials $\propto \sqrt{\text{thickness}}$, generally (for Para rubber, strength $\propto$ thickness). (Approx. values below.)

Material.	Volts for 1 mm. thickness.	Material.	Volts for 1 mm. thickness.
Ordinary paper	1,500	Varnished paper and linen.	10,500
Fiber and Manila paper. . . .	2,200	Ebonite.	28,500
Presspahn and Impregnated paper.	4,500	Rubber	21,000
		Gutta-percha.	19,000
		Para rubber.	15,500

(C. Kinzbrunner, *Electrician*, London, 9–29 and 10–6–'06.)

Electro-Magnets, Table for Winding.

Size of Wire, B. & S.	Single-covered, Turns per In.	Single-covered, Turns per Sq. In.	Double-covered, Turns per In.	Double-covered, Turns per Sq. In.	Size of Wire, B. & S.	Single-covered, Turns per In.	Single-covered, Turns per Sq. In.	Double-covered, Turns per In.	Double-covered, Turns per Sq. In.
4	4.73	26.1	4.58	24.5	18	22.08	568.7	19.88	461.1
5	5.29	32.7	5.11	30.5	19	25.07	733.3	22.8	606.5
6	5.92	40.9	5.68	37.7	20	27.81	902.2	25.03	730.9
7	6.61	51.	6.32	46.6	21	30.81	1107.6	27.41	876.6
8	7.55	64.2	7.18	60.1	22	34.07	1354.3	29.98	1048.4
9	8.24	79.1	7.81	71.2	23	37.64	1652.8	32.68	1245.8
10	9.18	98.3	8.63	86.9	24	41.49	2008.2	35.59	1477.7
11	10.44	127.2	9.88	113.8	25	45.66	2432.4	38.6	1738.2
12	11.65	158.3	11.01	141.4	26	50.15	2933.8	41.77	2035.5
13	13.	197.1	12.21	173.9	27	54.95	3522.9	45.04	2366.4
14	14.48	244.6	13.5	212.6	28	60.1	4213.	48.45	2738.4
15	16.11	302.9	14.8	255.5	29	65.57	5016.2	51.96	3149.9
16	17.92	374.7	16.44	315.3	30	71.27	5926.1	55.47	3589.5
17	19.9	461.9	18.26	388.9					

'Turns per sq. in." are calculated on the assumption that the number of layers per in. depth = No. of turns per in. (linear) $\times$ 1.166 (or $16\frac{2}{3}\%$ increase per in. due to imbedment of layers), and that "Turns per sq. in." = $1.166 \times (\text{turns per in.})^2$.

No. of feet of wire in 1 cu. in., L = Turns per sq. in. $\div$ 12.

Ohms resistance per cu. in. = $L \times$ No. of ohms per linear foot (see table on page 155).

Insulation assumed, δ (diam. of covered wire = diam. of bare wire + δ):

Size of Wire,	4 to 10 inclusive	11 to 18 inclusive	19 and up
Single-covered, δ =	0.007 in.	0.005 in.	0.004 in.
Double-covered, δ =	0.014 in.	0.010 in.	0.008 in.

E. M. F. of Dynamos. Let $2p$ = No. of poles, $2a$ = No. of parallel armature branches into which the current divides; then, $E = \Phi a n o \frac{N}{60} \frac{p}{a} 10^{-8}$. Let $\alpha = \lambda \div \tau$ (λ = pole arc, τ = polar pitch), B_l = induction in air-gap, D = diam. of armature in cm., l = length of armature in cm. Then, kilowatt capacity of generator = $clND^2 10^{-6}$, where $c = \alpha B_l A 10^{-5} \div 6$. ($A$ = No. of ampere-conductors per cm. of circumference, $= n o I_a \div 2\pi D$, where I_a = amperes in each conductor) A (ordinarily = 200) may reach 300 to 350, with high B_l, strong saturation of teeth and good ventilation. (If $\alpha = 0.6$ to 0.85, B_l = 6,000 to 10,000, A = 150 to 200, then c = 1 to 3.) The current volume in one slot of an armature ($= I_a n o$) should not exceed 900 amp.

If $I_a < 70$ amp., round wire should be used; if >70 amp., conductors of rectangular section are preferable. No. of commutator segments $= 0.04 n_o \sqrt{I_a}$. For n_o see bottom of page 136.

Current density in armature conductors: 2 to 5 amp. per sq. mm.(=400 to 1,000 cir. mils per amp. = 1,300 to 3,200 amp. per sq. in.).

Tooth saturation: maximum (at root) = 16,000 to 23,000 lines per sq. cm.; minimum (at periphery) = 14,000 to 20,000.

Saturation of core: 7,000 to 12,000,—lower value for multipolar machines.

For cooling of armature allow 5 to 10 sq. cm. of external surface for each watt wasted. (Kapp.) Brushes: each metal brush should cover from 1 to $2\frac{1}{2}$ commutator segments (carbon, 2 to $3\frac{1}{2}$).

Interpoles, Motors and Generators with. Interpoles are used between the main poles of multipolar machines for the purpose of neutralizing the armature magneto-motive force and the reactance voltage due to the short-circuiting of the armature coils by the brushes, sparking being thereby reduced to a minimum. The higher the speed, the voltage, and the output, the greater are the advantages derived from their use. Roughly, for generators,

K.W.	Voltage.	R.P.M.	Interpoles are:
750 and up	250 and up	1,500 and up	To be used.
250	250	1,000	Of slight advantage.
100	250	1,000	" " "
100 and up	250–500	100	" no "
400 " "	500	200	To be used.
600 " "	250	200	" " "

In the second and third cases, interpoles are more satisfactory, but they increase cost of construction, and good designs are available without using them. Interpoles are extensively used in small motors and dynamos of high and moderate speeds, but where heating and not sparking is the limit of output, their use is attended with increased cost, lowered efficiency, and no especial advantages.

The peripheral speed of commutator should not exceed 115 ft. per sec., and commutator should be large enough to radiate the heat generated, 1 sq. in. of surface being allowed for each 60 amperes of current taken off.

The leakage or dispersion coefficient is larger than in designs without interpoles, being 1.35 for the main magnetic circuits and 1.45 for the auxiliary or interpole circuits.

To calculate the flux required to enter the armature from the interpoles, let λ = length of conductor (in cm.) which actually cuts the auxiliary field. Then, $\lambda = 1.1 \times 0.7 \times b$, where b = breadth of pole-shoe (|| to shaft), 1.1 = coefficient to allow for "fringing" or spreading of field at the pole-tips, and 0.7 = that portion of the length of conductor which is active (i.e., imbedded in the armature iron, the remaining 0.3 being taken up by airducts, insulation, etc.).

Let S = peripheral speed of armature in cm. per sec., and B = average density in the air-gap of interpole in lines per sq. cm. Then, E.M.F. generated by one conductor $= B\lambda S \cdot 10^{-8}$. As there are two conductors in the short-circuited turn, E.M.F. in one turn $= 2B\lambda S \cdot 10^{-8}$, and this must suffice to neutralize the reactance voltage. If v = mean reactance voltage [= reactance voltage $\div (\pi \div 2)$], $v = 2B\lambda S \cdot 10^{-8}$, whence B, or the desired flux density $= v \cdot 10^8 \div 2\lambda S$. See pages 140–143. (H. M. Hobart, *Elec. Review*, N. Y., 1–20–'06.)

Resistance of Iron and Steel Rails. Iron rails have x times the resistance of copper conductors of same cross-section and the content of manganese in the iron seems to be the chief factor in increasing the value of x. For continuous currents, $x = 5 + 7$ Mn (roughly), where Mn = per cent of manganese. A very good rail used in London and containing 0.19% Mn has a measured value of $x = 6.4$. (By formula: $x = 5 + (7 \times 0.19) = 6.33$.)

Superheated Steam. The latest and most accurate data on the specific heat of superheated steam at constant pressure (k_p) are those obtained by Messrs. Knoblauch and Jakob, in 1905, at the Munich Imperial Technical College. They found that at saturation k_p rises very rapidly with the pressure: but as the temperature rises above that of saturation, k_p at first rapidly falls, reaches a minimum, and then rises more slowly with further increase in temperature; also that the temperature giving the minimum becomes greater with greater pressure. At higher temperatures the specific heats at all pressures approximate each other much more closely than at lower temperatures. The following table embodies the results obtained by these experimenters:

Absolute Pressure, lbs./sq. in.	Specific Heat (k_p) at Saturation.	302° F.	392° F.	482° F.	572° F.	662° F.
28.44	0.480	0.477	0.472	0.473	0.478	0.493
56.88	0.512	0.510	0.492	0.483	0.486	0.497
85.32	0.548	.....	0.513	0.491	0.490	0.500
113.76	0.582	.....	0.538	0.499	0.494	0.503

The mean specific heat through the whole range of temperature from saturation up to the various pressure-temperature conditions given is as follows:

Gauge Pressure.	300° F.	400° F.	500° F.	600° F.	700° F.
50 lbs.	0.525	0.507	0.497	0.494	0.494
100 "	.....	0.560	0.528	0.515	0.512
150 "	.....	0.618	0.560	0.533	0.526
200 "	.....	0.692	0.585	0.548	0.536
250 "	.....	.....	0.625	0.570	0.548

These values have been obtained from diagrams plotted by Mr. Robt. H. Smith (*The Engineer*, London, Aug. 23, '07) from the data of Knoblauch and Jakob's experiments.

Stresses in Rotating Disks. Let D = outside diam. in ins.; d = diam. of hole in ins.; v = rim velocity in ft. per sec.; w = wt. of 1 cu. in. of metal used; f_t = tensile stress induced by centrifugal force. Then for a plain disk, $f_t = 4wv^2(D^2+Dd+d^2) \div gD^2$. For a solid disk ($d=0$), $f_t = 4wv^2 \div g$. For a conical disk, $f_t = 2wv^2(D^4 + 3d^4 - 4Dd^3) \div gD^2(D-d)^2$. For solid conical disk ($d=0$), $f_t = 2wv^2 \div g$. For disks such as are used in high-speed steam turbines, and which have logarithmic profiles whose equation is $y = a \log (x \div b)$, $f_t = 1.5\, wv^2 \div g$ when $a=b$; $f_t = 1.2\, wv^2 \div g$ when $a = \frac{2}{3}b$. (From article by A. M. Levin, *Am. Mach.*, 10-20-04.)

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